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"HONORED NAMES IN MARINE ENGINEERING."*

AN ADDRESS BEFORE THE AMERICAN SOCIETY OF NAVAL ENGINEERS FEBRUARY 23, 1927,

BY WALTER M. MCFARLAND, MEMBER.

In ancient days, among the Vikings, the great sea fighters of their time, it was the custom at their banquets to listen to the recital by the bards or skalds of the deeds of famous men who had preceded them, thus keeping the memory of these worthies alive with the younger men. The printing press among other things has helped to do away with this custom and we now look to the encyclopedias and biographies for this information. It has seemed to me, however, that it would be suitable, occasionally, at gatherings like this, to have one of the older men take up the work of the bard and tell, possibly from personal acquaintance, something that might be interesting about those who were the leaders in their

*Address delivered at the Annual Banquet of the American Society of Naval Engineers, February 23, 1927, at Washington, D. C.

day, and mainly to throw some light on their personality, and perhaps, by some anecdote, impress it on the memory of some younger men, thus enabling the maintenance of the tradition.

The picturesqueness of the ancient gathering is, of course, absent and the narrator of this evening is very different from the ancient minstrel with his poetic song, but let us hope that we may do something towards accomplishing the same result.

Besides the loss of the picturesqueness and color of the ancient feast there is another loss which causes real envy of the freedom of the olden time. While listening to the bard, the hearers were not in danger of dying of thirst and indeed antiquarian research is said to have disclosed that the practice of the listeners is set forth in an ancient poem which is an anticipation of the modern one, so popular before 1918—"Landlord fill the flowing bowl." The bard himself was allowed "time out" to renew his inspiration.

In our attempt to carry out a variation of the ancient custom under such greatly changed conditions, let us consider a few of those who have placed their names on the honor roll of marine engineering. It is fitting therefore that we should begin with the father of engineering in our Navy, Mr. Charles H. Haswell.

Haswell was the first Engineer of our Navy, the first Chief Engineer of our first regular war vessel, and the first Engineer-in-Chief of the Navy. He was responsible for the organization of the Engineer Corps as it existed for many years and substantially as it remained until the amalgamation with the Line in 1899. He was appointed as a Consulting Engineer to the Board of Naval Commissioners February 19, 1836, and on July 12th of the same year was appointed Chief Engineer of the *Fulton*, the first regular steamer in the United States Navy. After extremely important and valuable service to the Navy he left it in 1852. His great successor, Isherwood, nearly half a century afterwards, in writing about him and his connection with the Navy, said that the Engineer Corps was particularly fortunate in having as its first member and organizer such a man as Mr. Haswell who, in addition to his ability as an Engineer, was an accomplished gentleman who had the respect of all with whom he was brought in contact. The scheme of my task, which includes a number of names, forbids going into great detail about any one of them, so that many important and

interesting things about each must be omitted. Mr. Haswell continued in the active practice of his profession until he had attained the very great age of 98, retaining to the very last his mental faculties and also his physical activity. He was a wonderful mixer and extremely popular with his associates. All the younger men counted it an honor to be presented to him and whenever he attended a meeting of any of the societies he always held a reception. As an indication of his mental vigor to the very end the following incident is a fine illustration.

The Society of Naval Architects at its meeting in 1906, when Mr. Haswell was past 97 years of age, appointed him Chairman of a committee to draw up resolutions of thanks to Secretary Root for the interest he had shown in the advancement of the American Merchant Marine. I was a member of this committee, which included a number of others who believed themselves competent to express such resolutions in very creditable English. Mr. Haswell asked the rest of us to prepare the resolutions and then let him see them. We expected that he would simply look over what we had done and approve them. Instead of this he proceeded to read the resolutions carefully, drew a pencil through phrases in several places and wrote in new ones. When he had finished he showed it to me and said, "Do you not think that these are improvements?" They were improvements in every case. Just think of the marvel of this almost centenarian with the ability to improve the work of a committee of really able men! I am glad to say that I had the thought to save the sheet with his corrections. We had a new and revised version prepared and submitted to the Society, and I told the story as I have just told it to you. Of course, it was highly appreciated, as was my suggestion that the annotated report should be placed in the archives. He died the next Spring within ten days of his ninety-eighth anniversary. When his obituary was prepared, this story was included and the document with his corrections, accompanied by this extract from the obituary, is now framed in the office of the Society of Naval Architects.

At the first banquet of that Society in 1893 Mr. Haswell was called on for an address. He was then past 84. He told an amusing story in connection with his experience while Chief Engi-

neer of the Allaire Works in New York City before entering the Navy, somewhere about 1834 or 1835. Work was very slack and to keep the force occupied Mr. Haswell suggested to Mr. Allaire, the President, that they should build a tug and trust to selling it when completed in the Spring. Mr. Allaire's answer was, "Charlie, we never could sell it; there are two tugs in the harbor now." I heard him make an address at a banquet when in his ninety-seventh year which was one of the striking features of the evening.

My first personal contact with him was at a Congress of Marine Engineers and Naval Architects in London in 1897. He had come over from home alone although in his eighty-ninth year. We happened to stay at the same hotel and exchanged calls and began a friendship which lasted until his death, and which has always been a source of great pride to me as giving me the honor of the friendship of this wonderful old man. At the meeting of the Congress, the Secretary, whom I happened to know personally, asked me about the names of some of our Americans who ought to be called on and I told him they certainly ought to call on Mr. Haswell who was then the oldest engineer in the world in the active practice of his profession. Accordingly, he was called on, but unfortunately I had not had the time to warn him of it, so that while he made a few pleasant remarks he asked to be permitted to submit some reminiscences in writing which would cover the whole period which the Congress was to celebrate, namely, the sixtieth anniversary of the accession of Queen Victoria. I was the one next called on to speak and drew the attention of the audience to the wonderful thing that had just happened in their presence, in having an address from a man who had seen the first commercially successful steamer, the *Clermont*, and had traveled on it as a boy, who had also been on board the first steam war vessel, the *Demologos*, and who later on had been the first Chief Engineer of the first regular war vessel of the Navy and the first Engineer-in-Chief of our Navy. Having traveled on the first commercial steamer, the *Clermont*, he had lived to travel on the latest type of naval architecture, the *Campania*. It is needless to say that, when they knew about him, the audience was intensely interested and after the exercises had concluded, when I was standing near him,

a number of my friends came up and asked to meet him. This started the line and he held a reception at which everybody present was introduced. From this time on our British friends could not pay him too much attention, and the beautiful thing about it was that he could accept such attention and carry on to the delight of all. Truly a great name to head our honor roll.

The next name we shall consider is Isherwood, the famous Engineer-in-Chief of the Navy during the Civil War and famous in engineering history for his experimental work, especially with expansion of steam. Handsome, able and accomplished, he was a striking figure in any gathering. Circumstances compelled him to be a strong controversalist and his extreme tenacity of purpose and great ability made him a dangerous opponent. Such a man was sure to make enemies and after the close of the Civil War, with the change in the Department which came about with the new Cabinet under Grant, Isherwood was sent, after the completion of his term as Engineer-in-Chief, to exile at the Mare Island Navy Yard. There was not much routine work in those days and he availed himself of the opportunity and carried on some of the finest experiments which he ever made, the ones on screw propellers. Later work, with finer apparatus and better methods of recording data, have, of course, superceded what he then did, but it is very interesting to remember that as late as 1900, about thirty years after these experiments had been performed and reported, they were used by one of the ablest engineers of Great Britain as the foundation for his work in connection with screw propeller design.

He was an intensely practical man with strong common sense and I have often thought of his rather brutal but effective method of handling a situation which arose during the war. A gunboat had been completed, whose services were very much needed, and it was found impossible to turn the engines with full pressure of steam admitted to the cylinders. He sent word to the New York Navy Yard, where the vessel was located, to supplement the steam pressure with hydraulic jacks and to keep the shaft turning. This, of course, was very slow work at first, but gradually the speed increased slightly and the work was continued until the jacks could be dispensed with and then the process was continued until a

reasonable number of revolutions could be obtained. I heard him tell why he did this. His judgment was that the lignum vitae bearings for the propeller shaft had not been properly soaked before being bored out for the shaft, and so, when the vessel was in the water, they had swollen and gripped the shaft as a most effective brake. There was not time to put the ship in dry dock to remove the bearing and bore it out. His method, though very brutal, was very effective.

The first time I had any personal contact with him was when he had been retired a few years and I was detailed to prepare some information for him with reference to some of the then new vessels. He explained to me just what he wanted and dropped in occasionally while I was preparing it. At that time, although about 65, he did not wear glasses and had never done so. A year or so later I met him again and he was then wearing glasses and when I spoke to him about it he said he had done it at the suggestion of friends, although he could not see where they were giving him any particular benefit.

About 1911 a party of half a dozen of us called on him at his home one afternoon to pay our respects. The formerly erect, handsome and vigorous man had changed to an old, feeble and bent one, almost 90 years of age, but when he sat down and assumed a position which kept the upper body erect, his eye flashed as brightly as ever, and the conversation we had with him showed that his mental faculties were as keenly alert as ever. I had read somewhere that, while he was on duty on the Asiatic Station before the Civil War, he had been very sick and the doctors had given him up and told him that he must die. He replied to them that he was not going to die but was going to get well. This, of course, was a very good story, if true, so I took advantage of the opportunity to ask him about it. He said that it was an actual fact, and it seems to be an illustration of the effect of a powerful will over one's physical condition.

The two names that we have just considered are to all, except a very few present, only names of famous men whom they had never seen. The next name that we shall take up is of a wonderful man, who was personally known to a great many here, and was like a father to those of us who had the privilege of working for

him. You will all know at once that I refer to that lion-hearted engineer George W. Melville. The story of his career is a romance and one to thrill the reader who examines his record, so that its impression on those who served with him is to recall many days of enthusiasm and devotion. You all know the old saying that "No man is a hero to his valet," meaning, of course, epigrammatically, that no man is a hero to those who see him every day. Melville was the absolute disproof of this saying, because the better one knew him, the greater was their admiration.

Two years ago in an address presenting a statue of Melville to the Society of Mechanical Engineers, on behalf of more than one hundred of his admirers, I said:

"The statue of Admiral Melville which is before us clearly shows the leading elements of his character which went to make him one of the great leaders of our profession. These elements I should list as great will power, tenacity of purpose, absolute fearlessness and leadership. Indeed, many people who knew him in life and others who have looked at his pictures have said that he was really one of the old Vikings, who had appeared a thousand years after the rest had gone."

His courage was demonstrated from the very beginning of his naval career. He feared absolutely nobody and, when he had his mind made up, would go ahead regardless of consequences. His remaining alone in the engine room of the *Wachusett* when she started out to ram the privateer *Florida* was one illustration, and the record of his heroism in the Arctic on the *Jeannette* Expedition is one without a superior.

He was appointed Engineer-in-Chief of the Navy in the Summer of 1887 during the recess of Congress. His first report was quite a shock to naval traditions because he spoke fearlessly of conditions as they existed, with his recommendations for remedy and improvement. His name had not yet gone in to the Senate for confirmation so that it would have been very easy for the President to revoke the appointment if he had desired to do so. The report gave offense to certain officers in the Service and they managed to get a protest against Melville's appointment before the President, accompanied by a copy of his report. Cleveland,

however, was a man of the same mould as Melville, rugged, straightforward and absolutely fearless. When he had read the report he said, "I only wish that we had more such men as Chiefs of Bureau."

Some two years later a naval officer, who is now dead and whose name need not be mentioned, wrote a letter to a friend who was editor of a newspaper, criticizing very severely engineering in the Navy and in such a way that it was beyond the bounds of propriety. He gave the editor liberty to print it provided he printed it absolutely as it was written, without any change whatever. The letter was printed, with the postscript showing the authority, and somebody who saw it sent a copy to Melville. He felt that this was a good opportunity to let people understand that, when they undertook to criticize the Corps and the work of which he was the head, they must expect to take the consequences. He accordingly reported the matter to the Secretary of the Navy, who at once saw that the proper course was a court-martial. The officer, who was in command of a vessel, was detached and placed under arrest pending the meeting of the court-martial to come a few weeks later.

It seems that the editor was also a lawyer, and he inquired of the Navy Department whether a civilian witness could be compelled to appear before a Naval court-martial. At the time there was no authority to compel such attendance and he was so advised. Accordingly, when the court-martial was held, while there was no question about the subject and nobody doubted that it was an accurate reproduction of the letter, the technical details of proof required the production of the letter or proof that it had been sent. The editor refused to appear and the verdict was "not proven because of inability to compel the attendance of the witness." The effect of this action by Melville was most salutary and when there was any future criticisms they were couched in proper language.

It is hardly necessary to say much about his wonderful engineering ability because there are still in service many of the vessels the machinery of which was designed by him and which proved so remarkably successful. He seemed to have what might be called engineering intuition which amounted to genius and

always seemed to know just what to do to remedy any trouble. On a smaller scale, but where this same quality was absolutely vital to success, we have the story in Captain De Long's ice journals of the *Jeannette* Expedition where he speaks again and again of Melville's mechanical ingenuity under the trying circumstances of that isolated craft frozen in the Arctic ice. These ice journals also tell stories of his wonderful heroism, particularly the landing on Bennett Island, where De Long comments upon the tremendous difficulty of this work as follows: "Near the island the ice was all alive and Melville left his boat and supplies and carrying only a day's provisions and his instruments, at the risk of his life, went through the terrible mass, actually dragging the dogs which, from fear, refused to follow their human leaders. If this persistence upon landing upon this island, in spite of the superhuman difficulties he encountered, is not reckoned a brave and meritorious action, it will not be from any failure on my part to make it known."

An amusing story occurs to me as showing his quickness of decision. A bottom blow valve, fitted to a casing through the bottom of the ship, had got loose on the stem and could not be operated. The circumstances were such that the vessel could not be put in dry dock so Melville determined to overhaul it anyhow. He prepared a large plug, to fit the opening when the valve bonnet was removed, and had it securely fastened to a capstan bar with some men all ready to jam it into place when the bonnet was taken off and the valve lifted out. The theory was all right, but when the bonnet was removed, there was such an outrush of water that the men lost their heads and failed to get the plug in place. Until they could get things arranged again, Melville solved the difficulty by sitting down on the opening. I think it is safe to say that the vicinity was somewhat blue from his remarks at the failure of the men to carry out his plans.

With all his bravery and outspoken frankness he was a man of great affection, as all of us who were associated with him daily learned thoroughly. I shall never forget an example of it when I was about to resign from the Service while one of his assistants. He said to me towards the end of December of '98, "Has anything been said to you about a farewell dinner by the boys here

in the office?" to which I replied that there had not. He then asked whether I had not been rather surprised because it had been a custom when one of our number went away for the rest to give him a farewell dinner. I told him that I thought that probably it was because it was near Christmas and everybody was busy so that there was no opportunity. He said, "That isn't it, you are entirely wrong. They wanted to give you a dinner at one of the hotels, but I told them that I knew something better than that, and I wanted them all to meet you in my rooms the night you are going to leave Washington for a little supper and farewell gathering." The affair took place and was delightful in every way, and one that I shall never forget. He presented me with a beautiful loving cup from himself and all my other friends there. When somebody asked him humorously, "What is the capacity?" his answer was, "One case of champagne." I left to take the train and so do not know whether this amount actually was consumed, but I think it quite possible.

My relations with him in my last tour of duty had become almost like those of a father and son and this continued during the ten years of his life after his retirement. He came to Pittsburgh occasionally while I was there and when I went East to New York I used to see him often, either there or in Philadelphia. I know that the others here, who like myself had the privilege of serving under him, will never forget our affection and admiration for this truly great man.

I believe all of you who knew him as well as I did will think that I ought to say a few words about Nathan P. Towne who was Chief Designer at the Bureau during the early part of Melville's administration and was connected with such famous ships as the *Columbia, Minneapolis, New York* and *Brooklyn* and others. The Cramps, who had had constant dealings with him, realized his ability and when their Chief Engineer, Mr. Younger, died they persuaded Towne to resign from the Service and go with them, where he remained for the rest of his life.

Most of us will think of him not so much for his ability as for his quick wit and sharp tongue and repartee. They will also remember his constant addition, to any epigram which he delivered, of the phrase, "See Luke IX, 17." As he made this apply to all

sorts of diverse things and many of them very far indeed from religious matters, it occurred to somebody one day to ask him if he knew what it really referred to. He said he certainly did. Deacon Smith, who was Chief Clerk of the Bureau, was a very pious man and had a Bible in his desk. Towne went over with the rest of the party, the Deacon got out the Bible, turned to the text, and Towne was able to recite it word for word. I may remark, for those of you who have not looked it up lately, that it refers to the miracles of the loaves and fishes, particularly with regard to the gathering up of the fragments.

One of the stories that Towne told was, when he was a young man, about going with a friend to a revival service. The minister became very much wrought up and in his enthusiasm asked all who wanted to go to Heaven to stand up. Everybody stood up except Towne. It seemed incredible to the minister that anybody should not want to go to Heaven, so a little later, when the penitents had gone forward to the altar and were prayed with by some of the assistants, the minister went to where Towne was sitting and said, "My young friend, I noticed that you did not rise when I asked all to stand who wanted to go to Heaven. Surely it cannot be that you do not want to go there?" Towne's answer was, "Of course I want to go there, but I am paired with a man who knows that he is going to Hell. He isn't here, so I can't vote." Not long before Towne left the Bureau to go to Cramp's, Sir Alfred Yarrow, then plain Mr. Yarrow, was traveling in this country and spent several weeks in Washington. He used to drop in at the Bureau frequently and finally came in one day to say good-bye. He spoke of his itinerary, which included a visit to Philadelphia. Towne said to him, "Why do you want to go to Philadelphia?" Yarrow's answer was, "There are many interesting places there like Cramp's, Baldwin Locomotive Works and many others." Towne's reply was, "But do you know it is such a terribly slow place; everybody goes to sleep there every day from 12:00 to 2:00. Did you never hear of the tale of the old Quaker lady who said to her little son, "Samuel, what do you mean, disturbing the neighbors by dropping pins on the sidewalk!" I never heard his opinion of Philadelphia after he became resident there, but I imagine it changed.

Another name, of which all of us as engineers can be very proud, is that of Robert W. Milligan who was Chief Engineer of the *Oregon* on her famous trip from San Francisco to Jupiter Inlet and during the Battle of Santiago. It seems to me that this is one of the cases which ought to be praised to young men continually, as showing how fame may come from the conscientious and intelligent performance of duty without the man who did it having any idea that his conduct was at all unusual. While the *Oregon* was on the way around, Milligan wrote me a letter from Valparaiso which I received before the vessel got to Jupiter Inlet. He spoke of the excellent way in which the machinery was operating and remarked that he had been able to get along up to that time without putting anything but fresh water in the boilers. He said, "I am afraid that I may become the most unpopular man on the ship because, if necessary, I shall insist on everybody being put on an allowance of water in order that the boilers may have only fresh water." I learned later on that, when they coaled ship at Valparaiso, they were able to fill the bunkers with a very fine grade of Nixon navigation coal. When they got around to Montevideo, the ship was met by colliers from home with some of our own coal which had been purchased on the "lowest-bidder" principle. It was soon found that this coal was distinctly inferior to the Welsh coal. As a result, when the ship got to Santiago and was on the blockade, Milligan had all of the inferior coal cleared out of the fighting bunkers, had them filled with this high grade Welsh coal, and then locked the doors, he carrying the keys. He was preparing for the day of battle!

There seems to have been an impression on the part of nearly all of our officers that Cervera's squadron would never come out, and as a result nearly all the ships went to half power by letting fires die out in half of the boilers. Milligan, however, believed in being ready for full power if it was needed, and, when Captain Clark spoke to him about running with two boilers only, he expressed the opinion that it would be unwise to do so as this was the one time for which the ships had been built, and it would be too bad to have them ineffective if the emergency arose. A few weeks later, Captain Clark took the matter up again and told him that he had come to the conclusion that it was a waste of fuel

to go on with all of the boilers; that no other ship was doing so; and he thought that fires should be allowed to die out in half of them. Milligan said, "Of course, Captain if you give me the order I shall cheerfully obey it; but I would be glad if you would give it to me in writing and let me acknowledge it, with a respectful statement of my opinion that a mistake is being made, although, of course, I shall carry out the order." Clark then said, "Well, if you feel that way about it we will let things go on as they are." A great many of you know the story; how, when the signal came for full speed to engage with Cervera's fleet, the *Oregon* was one of the two ships which had all boilers in operation. Milligan unlocked the doors to the fighting bunkers, the blowers were started, and in a very short time the *Oregon* was going at full speed and beating her trial trip record. She actually led the van ahead of the *Brooklyn,* whose trial speed had been 21 knots against the 17 of the *Oregon,* but half of whose engines were unavailable because uncoupled, and half of whose boilers were not only without fires but actually empty and with manhole plates off.

This fine performance was evident to all because out in the open. Another of Milligan's meritorious actions was known at first-hand only on the *Oregon.* She had been under way, in the chase of the Spanish fleet for some time without firing any of the guns. The men of the Engineers' Force, down below, and many in the air tight firerooms, were straining every nerve and, as we know, getting wonderful results, but they could not know how the battle was going, whether to success or defeat. To encourage and stimulate them, Milligan asked Captain Clark to fire a few shots from the forward turret. This was almost over the firerooms and would be the best news to the men below, as showing that they were chasing the enemy. We are told that the men cheered when they heard the guns. The record of the *Oregon* is due entirely to the long and successful trip around through the Straits of Magellan and to her wonderful work on the day of battle at Santiago. This is so clearly due to Milligan's engineering skill and faithfulness that one would suppose Captain Clark would have been only too glad to give him credit. If his desire to reduce boiler power had been carried out, no one would now know who was the commanding officer of the *Oregon* any more than

they now know who was the Commander of the *Indiana*, although Harry Taylor was as fine an officer as we ever had in the Service. When his official report was made, Captain Clark said something like this, "The conduct of all the officers under my command has been so fine that it would be invidious to make any special commendation and I therefore give the list of all of them." Captain Clark was a fine man and justly popular and I have always felt sorry that his record is not perfect because of his failure to give Milligan the credit which was deserved.

There are three names that deserve to be held in loving memory by younger engineers, especially by those of us who were cadet engineers at the Naval Academy. John C. Kafer, David P. Jones and Charles H. Manning. There were others who were among our instructors, but these names stand out for their special ability and fatherly interest in the cadets, and the special part which they had in the securing of legislation authorizing the education of engineers at the Naval Academy and in the organization of the course of instruction. They were all men of ability and all made their mark in civil life after retirement as well as by their career in the Service. The Engineers' Club in New York is a monument to Kafer and the Amoskeag Cotton Mills in Manchester, N. H., may be considered as a monument to Manning. They were all men who stood up for the Engineer Corps and the proper recognition of engineering in the Service. At the same time they were all men of agreeable personality and were popular in all the various branches of the Service.

Then we come to a group of still younger men whose special distinction in the Service was their ability as Designing Engineers, Asa M. Mattice, Alfred B. Canaga, Frank H. Bailey and Emil Theiss.

Mattice left the Service at an early age and after a comparatively short service of some fifteen years, but he had left his imprint both in the organization and traditions of the Bureau and in work at the Naval Academy where he was an Instructor in the Engineering Department. He made a remarkable record in civil life as a consulting engineer, especially in the great machinery of the Calumet and Hecla Mine, and as Advisory Engineer to the Remington Arms Company during the Great War. He was also

Chief Engineer of the Westinghouse Electric Company and Westinghouse Machine Company, and of the Allis-Chalmers Company. He was one of the most talented and brilliant men with whom I have ever come in contact. His knowledge seemed almost encyclopedic and certainly in engineering matters it always seemed as if he knew just a little bit more about any given subject than anybody else. With it all he was wonderfully modest and almost retiring. Indeed, in his last years he almost withdrew from general society except to his most intimate friends. I well remember an article which I had prepared for one of the Societies when we were both connected with the Westinghouse interests and I had asked some of the other engineers to look it over to check any mistakes or to suggest any additions. Finally, I asked Mattice to look it over. His annotation was almost like a new paper. Those who knew him well will remember his collection of remarkable stories, but time is too short now for me to tell any of them.

Canaga and Bailey were able designers and are specially remembered for their remarkably even dispositions which it seemed no amount of nagging or worry could trouble. Both of them had the good fortune to be Chief Designer at the Bureau at a time when very important work was going through, so that they could feel that they had left a record which would last for many years. Theiss came some years after these others and is the only one thus far mentioned who was my junior. He was a man of fine ability and a worthy successor to these splendid men. I remember an incident, when we were both at the Bureau under Melville, at the breaking out of the Spanish War, and Theiss was the Assistant to the Chief Designer. He came into Melville's office while I was there and told him that he would like permission to apply for his detachment. "What is the matter," said the big Chief, "Don't you like the work here, haven't I treated you right?" "Yes," said Theiss, "I love the work and nobody could have been kinder than you, but I have been trained as a fighting man and this seems to be the only chance in my time to get into it so I would like to be detached and go to the front." "That is exactly right," said Melville, "and I would have a poor opinion of you if I didn't believe that you wanted to go. I would like to go, too, but some of us must stay here and carry on this work, which is just as important.

I could lay my hands on fifty men who could do the work at sea which would come to you, but I don't know where to turn to get a man to fill your place here."

Thus far the saga has been about Naval Officers, but there are big names in marine engineering in civil life and some of them were men whom I knew well and admired.

Charles H. Cramp was for many years the head of the great Cramp Shipyard and I frequently met him during his numerous visits to the Bureau while I was an assistant to Melville. I always enjoyed a chat with him, and can say very frankly that there was hardly a time that I did not learn something from this able and accomplished man. He was a master of the profession and had had a remarkable experience. He also had a very keen tongue and did not hesitate to use it. I remember, for example, that when the early ships of the new Navy were under contract on a horsepower basis and it was necessary to standardize the indicators carefully, Uncle Charley always referred to the "corrections" as "deductions." I was speaking to him once and referring to them as corrections, when he said, "No, no, don't say that; there are never any increases; they are always deductions."

The first trial of the *Baltimore* did not come up to their expectations and they were persuaded to employ an alleged expert with the indicator. In order to make a bluff at earning his money he made a gesture by submitting a lot of criticism of the method of standardizing the indicators used by the Bureau, and raised a number of points which were ridiculous, because they covered matters of so little weight that the variation in the thickness of the line of the card would be of greater importance. We answered the letter and said that, of course, if they insisted on all these points being taken into consideration, we would do so, but, of course, if such extreme refinement was to be considered, there would be other items to be taken into account, many of which would certainly work the other way, and in all probability the net result would be to make the corrections larger than before. We itemized the various things and included in them the term "parallax," referring, of course, to the error in reading the scale, but merely using the word. It is said that when Uncle Charley read the letter he commented, "What the Hell is 'parallax'!" To complete

the story it may be said that Cramps replied that they would waive the benefits which might be received from the granting of their request.

Some years after I had left the Service, I happened to see the manuscript of a brief autobiography of Uncle Charley in which he was doing something of the same thing that I am trying to do tonight, but with a little more ginger and vinegar in his remarks. In commenting on persons and places he referred to the Navy Department and spoke of various officials before the Civil War. One paragraph was about as follows:

"The Chief Constructor was a fine gentleman of the old school, very courteous and considerate; and also an accomplished Naval Architect of the old school. Unfortunately he thought that steam should be auxiliary to sail instead of the other way round. His assistants were opposed to everything and would do nothing themselves."

This last is one of the most remarkable epigrams that I ever read.

A younger man than Uncle Charley, and his rival, was Irving M. Scott, the Vice-President and General Manager of the Union Iron Works at San Francisco. Scott was a very able and agreeable man and all of us who used to meet him were very fond of him. His strong point was the splendid workmanship turned out on all the ships built by his firm. To this I can testify from having served on the *San Francisco* and it was also shown in the great success of the *Oregon*, the *Olympia*, and others. The special point which I remember about Mr. Scott is a clever bit of advertising which he did for his firm in connection with the bids for the cruiser *Newark*. When the contractors came to make their estimates and allow for a moderate profit, it was found that the appropriation was too small. This was a matter of common talk and it was expected that all the bids would exceed the appropriation. Scott came East from San Francisco a week or ten days before the opening of the bids and picked up all the gossip, including this point, with which, of course, the Union Iron Works were in agreement as their prices, as a rule, were higher than those of the yards on the East Coast. He called on the Judge Advocate General and asked him if the low bidder would get the contract

even if he exceeded the appropriation. The Judge Advocate General, patterning after the big courts, declined to answer it as being a hypothetical question. Then Scott went to see the Secretary, who answered that, of course, no contract could be awarded if all the bids were over the authorized price. It was necessary to put up a certified check for 10 per cent of the bid, so Scott telegraphed home to have such a check arranged for, and, to everybody's surprise, when the bids were opened, the one of the Union Iron Works was the lowest. He had assured himself that he would not have to take the contract at this figure because it was over the appropriation, and so, at the mere cost of the certified check, which was a small amount, he temporarily got the credit for having outbid the big Eastern yards.

In my knocking about the world I have come in touch with some of the foreign celebrities in our profession, and a word about one or two of them may be interesting. In 1897 I was detailed as one of the representatives of our Navy Department, the other being Admiral David W. Taylor, then an Assistant Constructor, to the Congress of Marine Engineers and Naval Architects, to which I have already referred when speaking about Mr. Haswell. I had just finished a cruise on the *San Francisco* and went home by way of London. During the previous Winter we had been moored for a few days in the Harbor of Naples and an English yacht came in alongside of us. One evening, when I returned to the ship after a visit to the shore. I learned to my surprise that Sir William White, the famous Naval Architect and Director of Naval Construction for the British Navy, was a guest on the yacht and had been over with the owner to make a call on us. I hoped to get a chance to see him before we separated, but it did not occur. The next Summer, when I went over to the Congress in London, I felt in duty bound to pay my respects to him and the Engineer-in-Chief, Sir John Durston, and called first on the latter. While I was in his office, Sir William White came in, whom I recognized at once from his pictures, although I had never seen him before. I was introduced and said to him that I was very sorry that I had failed to see him when he had been in Naples, the Winter before, and when I was on the *San Francisco* at the time he visited it. He said, "I wish I had known that you were there. If I had, I would

have made it a point to come over to see you." I mention this merely to give you an idea of his suavity, which some unkind people might call "blarney," because I am sure that he had never even heard of me until I was introduced to him. I met him a number of times afterwards and we became very good friends.

I was in England again in 1911 at another of these Congresses, this time as a delegate from the Society of Naval Architects, of which I was then one of the Vice-Presidents. Sir William White presided over the Engineering Section of the Congress. One of the papers was by Sir Henry Oram, the Engineer-in-Chief of the British Navy, and in it he called attention to the change that had been made in their organization, which was quite similar to ours as respects the amalgamation of the Line and Engineers, and mentioned that the last of the engineers of the old school had entered the year before. Among other speakers who discussed the paper was Lord Charles Beresford, who has often been referred to as the "Bob" Evans of the British Navy. This designation I think was quite accurate, as both of them were great advertisers. He made a very pleasant little speech, and, among other things, expressed his great regret at the fact that the engineers of the old school were about to disappear. He spoke of having had many of them under his command and they had all been faithful and efficient. It was altogether a very complimentary statement. A little later Sir William White looked down at me in the audience and asked if I did not want to say a few words. The spirit moved me and I went up to the platform. I first paid my respects to Admiral Oram, whom I knew slightly, and about whom I had known for years on account of his being a classmate at the Royal Naval College of Gatewood and Bowles, who were my classmates at Annapolis; and I spoke of the interest I had always taken in his work and the commendation which I could certainly give to his official career. I then said that I wanted to express my appreciation of the highly complimentary remarks made by Lord Charles Beresford with respect to the engineers of the old school, and, on behalf of these engineers, of whom I was one, although in another Navy, I wanted to express my thanks to him; and I would only add that, if, in times past, such expressions had been more frequent, there would have been much less hard feeling and

contention in the navies of the world all around. When I returned to my seat in the audience, a man leaned over from behind me and said, "I take it from your remarks that this is about the first time that you had ever heard of any such commendation."

It will occur to all of you that there are names of men in the old Engineer Corps and later as well as in Civil Life which might have been mentioned; and you must all have noticed, of course, that no names of living men were touched upon. Even as it is, my remarks have taken too much time, so you can see that to have included other names would simply have made this address impossible. In the beginning, I said that I would try to make this little address somewhat analogous to the songs of the bards at the Viking feasts, and mentioned some of the points of difference. There is another one. Those feasts used to last all day, and the bard was the only one that had anything to say, so far as we know. If a man had five or six hours to fill up, he could mention a great many more people than when his time is limited to thirty minutes. I am grateful to you for your kind attention.

NAUTICAL CHART CONSTRUCTION.

By JOHN H. LARRABEE,
SENIOR ENGINEER, HYDROGRAPHIC OFFICER, U. S. NAVY.

While the use of nautical charts is familiar to all navigators and followers of the sea, it is doubtful if the many processes and details entering into the production of such charts is realized or appreciated by those who are so vitally dependent upon their use to safely guide their ships over the seven seas and from port to port throughout the world. It is the purpose of this article to acquaint the user of nautical charts with the various phases of their construction from the obtaining of original data upon which the charts are based down to the finished printed chart issued for use aboard ship.

The Hydrographic Office, U. S. Navy, is authorized by law to provide accurate charts for the use of all vessels of the United States, and for the benefit and use of navigators generally, thus providing for the improvement of the means for navigating safely the vessels of the Navy and of the mercantile marine. Other legislation gives to the United States Coast and Geodetic Survey the task of making original surveys and printing charts of the coasts of the United States and United States Territorial waters, leaving to the Hydrographic Office the corresponding work on the high seas and in foreign territorial waters.

So little is known generally about the Hydrographic Office of the Navy Department that a short description of its history may prove of interest.

In the earlier days of our naval activities no systematic provision was made for supplying our naval vessels with the necessary charts, navigational tables and manuals, navigating instruments, etc., that they required. When a ship was to be fitted out, these things were collected from here and there, by purchase from outside individuals, because the Government itself did not produce

any of these most necessary articles. At the end of a cruise, when a ship was to be placed out of commission, her charts, books, instruments, etc., were tumbled into storerooms at the place where the ship was to be laid up, and they remained there, with little care or attention, until they were needed for fitting out another ship, when they were in many instances found quite unfit for use.

With such conditions existing, in December, 1830, in order to eliminate the difficulties and dangers to which our naval vessels had previously been exposed from the lack of an orderly and sufficient supply of information respecting the navigation of those waters of the world into which the performance of their duties constantly called them, it was ordered by the Navy Department that a depot should be established at the seat of government for the purpose of taking charge of such nautical charts and instruments as had collected in the various navy yards, and of assuming the care and issue of charts and instruments to United States naval vessels.

The difficulties that were experienced in obtaining and maintaining an adequate supply of the latest charts early led to a recommendation from the Board of Navy Commissioners to the Secretary of the Navy that a means of producing charts should be installed at the depot, and the introduction of a chart printing press at that depot in May, 1835, constituted the initial attempt at chart production by what has since become the Hydrographic Office. With the decision to produce its own charts for our own naval vessels, the department took the steps necessary for surveying the areas for which such charts were most needed.

The practical difficulties in regard to housing the depot were subsequently met by the passage of an act to authorize the construction of a depot for charts and instruments of the Navy of the United States, on August 31, 1842.

From this beginning the Hydrographic Office elaborated its work to meet the demands made upon it, sometimes working under the cognizance of one bureau and sometimes under that of another, but always under the Navy Department. It gradually gained greater and greater recognition from Congress as the value and magnitude of its labors became more and more apparent, such recognition being at first in the form of special appropria-

tions, but eventually taking the form of the organic law of June 21, 1866, which established the office as a separate institution and separated it from the Naval Observatory where the astronomical work of the Navy Department and the care and supply of navigational instruments continues to be conducted.

Since its establishment the Hydrographic Office has steadily expanded its chart producing activities until it now publishes some 2900 individual nautical charts covering the navigable waters of practically the entire globe. Many of these charts are from accurate surveys made by the U. S. Navy while the surveys upon which a large portion are based must of necessity be of foreign origin. There are, however, vast areas as yet unsurveyed, along the coasts and within the territorial waters of countries which on account of financial or other reasons are unable to make surveys. Such areas if surveyed at all must be surveyed by the larger and richer nations. The United States Hydrographic Office has been endeavoring to do its share of this work.

The Division of Chart Construction of the Hydrographic Office produces all Hydrographic Office charts issued, and its work embodies three major activities, viz., hydrographic surveys, nautical chart construction or compilation, and reproduction of nautical charts.

HYDROGRAPHIC SURVEYS.

In the conduct of Naval Hydrographic Surveys this Division determines the areas to be surveyed, prepares specifications for the surveys, examines and adjusts all survey data received from the field, and smooth plots the same for the use of the cartographers in the construction of the nautical charts. The hydrographic surveys which have been made by the U. S. Navy and under the cognizance of the Hydrographic Office are distributed throughout a large portion of the world and date back to as early as 1837 when the survey of Georges Shoal and Bank was made by the U. S. brig *Porpoise*, schooners *Maria* and *Badassah*, Lieut. Charles Wilkes commanding.

The United States exploring expedition, 1838 to 1842, was the first expedition to leave the shores of the United States fitted out

and provided for by an act of Congress directing its undertaking. March 20, 1838, Comdr. Charles Wilkes was put in command by the Hon. Mahlon Dickerson, Secretary of the Navy. The fleet consisted of U. S. sloop of war *Vincennes*, U. S. sloop of war *Peacock*, U. S. brig *Porpoise*, U. S. *Relief* (store ship), U. S. *Sea Gull*, U. S. *Flying Fish* (formerly New York pilot boats).

The object of the expedition was to make surveys to determine the existence of doubtful dangers reported in the track of the United States trade, to make astronomical observations for locating shoals, islands, reefs, etc.; observations for terrestrial magnetism, variation of the compass, etc.; to instruct the natives of the islands visited in agriculture and horticulture and to encourage them to increase their output; to discover if possible a shorter route to China via the Sulu Sea.

The Narrative of the United States Exploring Expedition is published in five volumes and an atlas. The surveys by this expedition were practically all published as charts, some of which are still on issue.

To form some idea of the field of operation covered by this expedition it is necessary to know that the work was started on the coast of Brazil (Rio Janeiro), thence to Madeira, and later to Patagonia, the Antarctic Continent (Wilkes Land), up through the Pacific Islands, Sulu Sea, Philippines, Japan, China, Fiji Islands, Hawaiian Islands, Samoan Islands, Alaska, Straits of Fuca, and the Columbia River, etc.

One item, Fiji Islands, illustrates the results obtained in the various groups. The area of the submarine plateau from which the Fiji group rises is about 40,000 square miles. The land area of the 225 reef-bound islands comprising this group is about 8,045 square miles. The bank, islands, and reefs were first surveyed as a group, for geographic position, shape, heights, depths, outline, and relative positions. Special surveys were then made of the more important islands, harbors, channels, etc., and the variation of the compass determined and tidal and current observations taken.

The Perry Expedition to Japan, 1852-1854 was primarily of a diplomatic nature for the purpose of opening ports for trade with the United States and paving the way for future scientific research.

Such scientific work as could be accomplished was to be done by the naval officers of the fleet. Observations were made and data obtained relative to the cyclones and winds of the western Pacific and the Kuro-siwo or Gulf Stream of the north Pacific. A great fund of nautical information was acquired and sailing directions compiled and published. Special observations were made on the Zodiacal Light. The fleet consisted of U. S. steamers *Mississippi, Saratoga, Plymouth, Supply* (store ship), *Susquehanna, Caprice, Vandalia, Powhatan, Macedonian, Lexington*, and *Southampton.*

The narrative of this expedition has been published in three volumes. Of the many surveys made by this expedition that of the Gulf of Yedo was perhaps the most complete.

The North Pacific surveying expedition sailed out of the Chesapeake Capes, June 11, 1853, under the command of Commander Cadwalader Ringgold, United States Navy. The fleet consisted of the U. S. steamers *Vincennes, J. Hancock, Porpoise, F. Cooper,* and *J. P. Kennedy.* Madeira was the first stopping place. The Pacific was reached via Cape of Good Hope and Sydney. Ringgold was taken ill at Macao and returned to the United States under care. Lieut. John Rodgers assumed command August 11, 1854.

The object of this expedition was identical with that of the United States exploring expedition, with the exception that its field of operation was confined to the north Pacific Ocean. A great deal of work was done by this expedition in adjusting and correcting charts based on old surveys.

From the work of this expedition the United States Governmen was enabled to publish detailed coasting charts of the entire coast of Japan, the coasts and islands of the Bering Sea and Strait, and of the Arctic Ocean as far north as Wrangell Island.

The most extensive hydrographic surveying operations conducted by the U. S. Navy in recent years have been in the West Indies and Central America. Soon after the Spanish American War in 1898 the Navy undertook the survey of the important harbors and bays in the Island of Cuba, and later started a project which calls for a complete hydrographic survey of all the coastal waters of the entire island. About one-half of this project has

been completed and the work is still in progress. Another extensive survey now in progress is that of the Gulf of Venezuela and entrance to Lake Maracaibo.

As an example of the need of modern hydrographic surveys in Central America and South America, the Hydrographic Office charts of the Caribbean Coast of Colombia now on issue were compiled from Spanish surveys made prior to 1837 and along extensive stretches of that coast not a single sounding has been taken and added to the charts for over 100 years. The same is true to a greater or lesser degree in many other localities.

THE NAUTICAL CHART.

In order to indicate the relation between the hydrographic survey and the nautical chart the following definition of a nautical chart is given:

A nautical chart is a miniature representation upon a plane surface, in accordance with a definite system of projection, or development, and drawn to a scale suitable for its intended purpose, of a portion of the navigable waters of the globe, including the outlines of the adjacent land, together with the surface forms and artificial features that are useful as aids to navigation; and setting forth the depths of water, particularly when such depths constitute a menace to navigation, especially in the near approaches to the land, by soundings that are fixed in position by accurate determinations; and based upon information gained by hydrographic, aerial, or topographic surveys, or any combination of these.

From the foregoing definition it is evident that suitable surveys must precede the construction of a nautical chart. While this article is not intended to cover in detail the methods employed in making these surveys it may be of interest to note some of the phases of the survey work necessary.

First a suitable control, covering the area to be charted, must be executed. This control is generally a system of triangulation expanded from an accurately measured base line. Since the control furnishes the relative positions of reference points to which all other data of the survey is tied, and hence controls the accuracy of the finished chart, the necessity for care and precision in its execution cannot be overestimated.

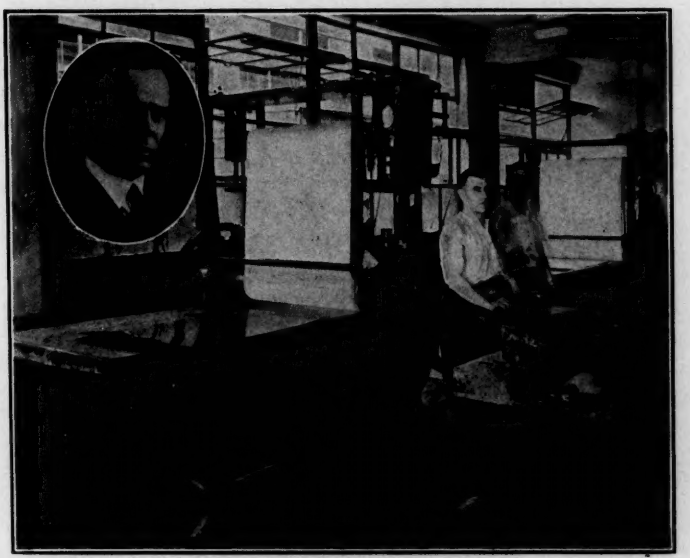


PLATE PRINTING PRESSES.
(Inset) JOHN H. LARRABEE.

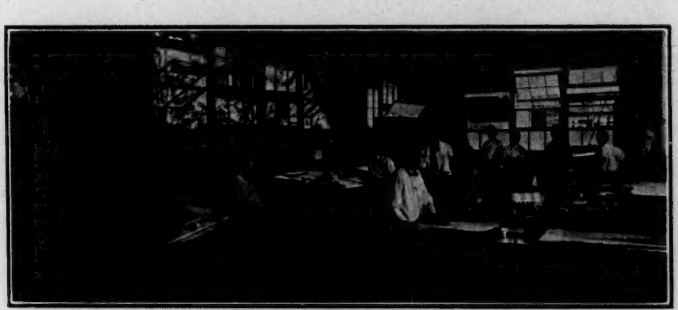
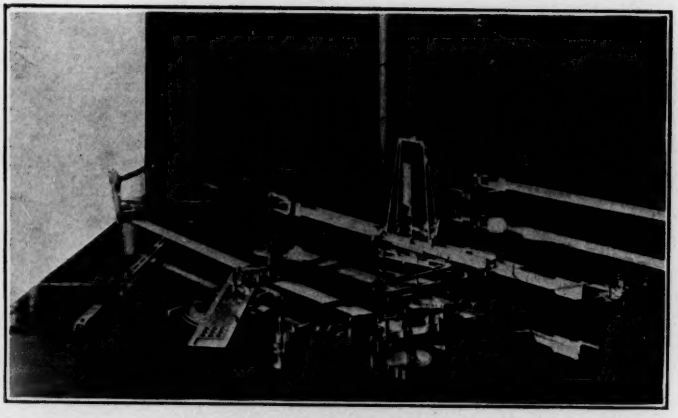
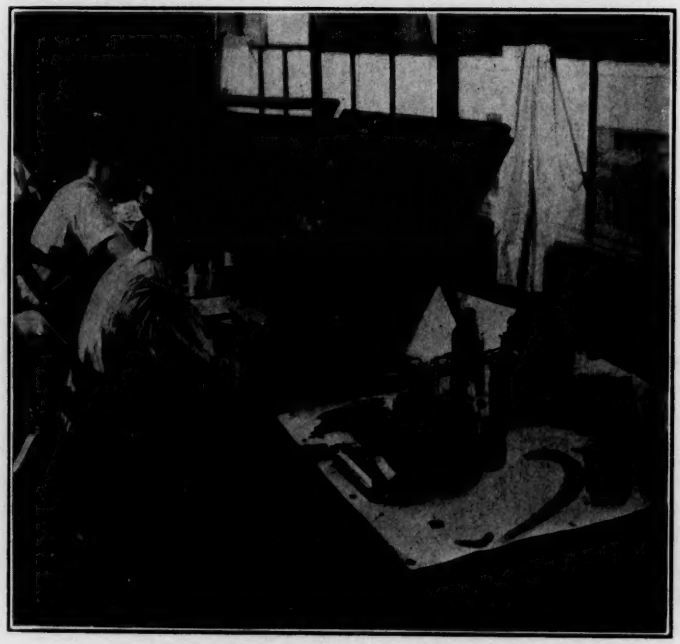


CHART DRAFTING ROOM
HYDROGRAPHIC OFFICE, U. S. NAVY.



THE "PANTOGRAVER."



NEGATIVE CUTTING.

As it is essential that the resulting chart should be correct in its geographic position a determination of the latitude and longitude of one of the triangulation stations must be made, and a true azimuth determined at the same triangulation station. From these data the geodetic coordinates of all the triangulation stations are computed.

Following the execution of the triangulation control a topographic survey is made of the shore lines, surface forms, and artificial features such as buildings, piers, breakwaters, aids to navigation, etc. Special care is taken to accurately locate all visible objects of a permanent nature which if charted might be of aid to the navigator.

The execution of the hydrography whereby the depths of water, character of bottom, and the location of shoals and channels is determined constitutes the major part of the survey work required to furnish complete data for the nautical chart.

PLOTTING THE SURVEY.

The surveys having been completed the next step is to smooth plot all of the survey data. A suitable scale for the smooth plotting sheet is chosen, usually an even natural scale, such as 1:20,000, 1:40,000, etc. (A scale of 1:20,000 signifies that one unit length on the plotting sheet represents 20,000 of the same unit on the earth's surface.)

A polyconic projection on the scale adopted and of sufficient extent to cover the area of the survey is laid down on the sheet. The control stations are then plotted in their computed geodetic positions. The sheet used for this purpose is of well seasoned, cloth-backed, drawing paper in order to insure a minimum change in the dimensions of the sheet during the plotting of the survey. Distortions occuring subsequent to the completion of the smooth sheet can be compensated for and eliminated from the final published chart, but distortions taking place during the plotting of the survey will result in an inaccurate chart.

After the control stations have been properly located on the sheet the topography is either plotted or transferred from plane table sheets depending upon the method of surveying employed. The soundings are then plotted and the depth contours or fathom

curves delineated. Although only a small portion of the soundings actually taken may appear on the final published charts, all of the soundings taken in the survey are plotted and utilized in the delineation of the fathom curves.

A smooth survey sheet does not necessarily cover the exact limits chosen for a particular chart. In fact many charts are made up from a combination of several survey sheets.

CLASSES OF CHARTS.

There are three general classes of nautical charts, viz., general or ocean charts, coastal charts, and harbor charts. A harbor chart is generally the result of a single survey; a coastal chart a combination of several surveys, while a general chart is a compilation in which many harbor and coastal charts have been combined and generalized.

The Hydrographic Office publishes a series of charts covering a large portion of the world on a standard scale of 1 degree longitude equals 4½ inches. These are general charts and are not intended for in-shore navigation. Hence the amount of detail of in-shore hydrography is minimized on these charts, only the principal lights are shown, and the nomenclature is confined to the more important localities and hydrographic and topographic features.

Coastal charts range in scale from about ¼ inch to the nautical mile up to about one inch to the nautical mile, while harbor charts are generally on a larger scale which depends upon the size, importance, and character of the harbor, as well as the completeness and accuracy of the survey. It is poor practice to construct a chart on a larger scale than that on which the survey is plotted, as in such a case enlarging the scale does not increase accuracy.

CHART REPRODUCTION.

The procedure for constructing a nautical chart depends to a considerable extent on the process to be employed in its final reproduction. From an article by the writer entitled "Modern Method of Chart Engraving" which appears in "The Military Engineer," Vol. XVII, No. 91, January, February, 1925, the following is quoted:

"In the early days of nautical chart reproduction, before the advent of modern lithographic and printing processes, it was the practice to reproduce nautical charts by engraving on copper or steel plates and hand-printing directly from the engraved plates. This practice was followed by the Hydrographic Office until 1912 when its first lithographic offset press was installed.

"With the development and perfection of photolithographic processes, the chart producer naturally turned to lithography as a means of reproducing charts more rapidly and more economically. Also, it was found that the lithographed reproductions of charts were superior to those printed directly from the engraved plates, in that the distortions found in plate printed copies, due to printing on wet paper, could be entirely eliminated in lithographic printing. The ability to print in various colors by lithography also enhanced its value as it provided a method of emphasizing distinctive features, a matter of great importance to the navigator in using nautical charts.

"From the foregoing, one would logically assume that the reproduction of charts by engraving and plate-printing would be abandoned in favor of the photolithographic process. Experience has shown, however, that because of the labor required in preparing a complete and perfect chart original for photolithography, and because of the necessity of continually altering and bringing up to date for frequent editions, the engraved copper chart plate provides the best original for photolithographic reproduction. The Hydrographic Office has, therefore, adopted the policy of engraving its nautical charts on copper as rapidly as possible and then reproducing by lithography, using the engraved plate as a base or original.

CHART ENGRAVING.

"The procedure involved in the hand-engraving of nautical charts is as follows:

"*Projections*—All nautical charts are constructed on computed projections. In engraving a chart on copper, the first step is to lay down the computed projection. This is done with great precision by the engraver, using a standard meter bar and rigid metal beam compass. Projection lines are then pointed in lightly on the

surface of the plate at intervals depending on the scale of the chart being engraved. This pointed in projection serves as a control for placing correctly the various features of the chart. As the engraving is made a reverse of the original, the projection must be laid down reversed.

"*Outlines*—The engraver is furnished with a drawing or photo print of the chart on a scale as near that of the engraving as practicable. Because of shrinkages and distortions in paper, the projections on the drawings or photo prints seldom correspond exactly to the computed projections laid down on the plate, and the engraver must adjust the work between projection lines in transferring the design to the plate. The features included under outlines are shore or coast lines, roads, railroads, buildings, rivers, boundary lines, canals, piers, docks, etc. The engraver carefully traces these features on a thin sheet of gelatine. The gelatine is tacked down over the drawing or photo print and the tracing done with a steel point which cuts into the surface of the gelatine.

"In addition to the outline features, the projection lines, fathom curves, limits of reefs and bluffs, positions of aids to navigation, topographic details and soundings, are also traced. After the tracing on gelatine is completed, sulphur dust (milk of sulphur) is rubbed into the lines cut by the steel tracing-point. The gelatine tracing is then placed face downward on the plate, the projection lines on the tracing superimposed on the corresponding projection lines on the plate, and the design burnished down by rubbing the back of the gelatine with an engraver's burnisher. The sulphur in the lines of the tracing produces a chemical action and causes the plate to discolor under the lines, thus producing a stained copy of the design on the surface of the copper plate. The plate is then ready for engraving. The outline features are cut with a hand-graver and the fathom curves are pointed in with a roulette. Aids to navigation and limits of reefs and bluffs are lightly pointed in. The plate is then ready for engraving of soundings, which is accomplished by the use of a machine designed for that purpose.

"*Lettering*—In the hand-engraving of nautical charts, the lettering of names, titles, and notes constitutes a large percentage of the work and requires the services of highly trained, skilled engravers.

Many years of experience are necessary for an engraver to acquire the manual skill required in engraving good lettering on map and chart plates.

"The lettering to be engraved on a chart is indicated on the drawing or photo print furnished the engraver with the names, notes, and titles correctly spelled, and in the approximate positions required. The gauge and style of letters to be used is also indicated.

"The engraver first places gauge lines on the plate in proper positions for all lettering on the chart. These gauge lines are lightly ruled into the surface of the copper with steel-pronged letter gauges, leaving lines which must be removed after the lettering is completed.

"The next step is to lay out and point-in the lettering. This is accomplished by sketching and scratching lightly the outlines of each letter on the surface of the copper plate with a steel point. The work is all backwards and requires great skill and adaptability in the proper formation of letters, and in centering lines of symmetrical notes or titles.

"After being pointed-in on the plate, the lettering is ready for engraving. The body strokes of the letters are engraved first, using a graver making a flat cut of the exact width of body line required. The body strokes must be 'returned,' that is, cut in the opposite direction from which the first stroke was made. The hair-lines are cut next, using a graver making a fine, narrow cut. The hair-lines also must be 'returned.' The letters are then 'trimmed' or 'finished' to blend properly the hair-lines into the body lines.

"*Topography*—Under the head of topography are included such features as bluff symbols, reef symbols, sand beaches, marsh symbols, symbols representing culture of various kinds, contours, form contours, and hachures. These features are engraved, some with a graver and some with a point, according to their character.

"*Fathom Curves and Sanding*—The fathom curves, having been pointed in with an appropriate roulette, are engraved by punching dots already pointed in for spacing. In order to emphasize the fathom curves, they are sanded back, that is, consecutive rows of

dots are engraved back of the fathom curves with spaces between dots progressively increased as their distance from the fathom curve increases, thus producing a blending off effect.

"*Aids to Navigation*—The engraving of buoys, beacons, lighthouses, and light sectors is left until the plate is nearly finished, in order to avoid erasures and corrections due to possible changes in aids to navigation during the time the plate is being engraved. The symbols for aids to navigation are usually made with steel punches, with appropriate design, which are held on the plate in proper position and subjected to a blow sufficient to impress them to a proper depth within the surface of the plate."

Within the past three years a mechanical instrument for engraving chart plates has been developed in the Hydrographic Office. This instrument known as the "pantograver" is the result of inventions by two employees of the Office. The pantograver is constructed on the pantograph principle and is employed in the engraving of outlines, soundings, and lettering on chart plates. An attachment to the pantograver automatically compensates for any distortion existing in the data sheets from which a chart is engraved. From seventy to ninety per cent of the work on an engraved chart plate can be produced with the pantograver and the remainder engraved by hand.

CONSTRUCTION OF THE CHART.

The limits and scale of a nautical chart having been decided upon the first step in its construction is the computation of a suitable projection. Nautical charts are generally constructed on a Mercator projection which is particularly adapted to navigation as a ship's track, with constant bearing, is a straight line on the chart. Also, the meridians being parallel as well as the parallels of latitude, a ship's geographic position is readily plotted on, or read from, the chart by means of latitude and longitude scales formed by a subdivided border around the chart.

On a Mercator projection the longitude scale is constant throughout the chart while the latitude scale varies, increasing in value as the latitude increases. Hence, no fixed scale can be applied to the correct measurement of distances over all positions of a chart on a Mercator projection.

In the case of harbor charts which are on a large scale and cover a comparatively small area the variation in scale throughout the chart is so small as to be negligible and fixed graphic scales are placed on the chart. These graphic scales are usually scales of yards. In addition, scales of meters are also placed on harbor charts for the convenience of navigators who are accustomed to using the metric units. The natural scale also noted on harbor charts is for practical purposes correct for all portions of the chart.

In the case of smaller scale charts, viz., coastal charts and general charts, the variation in scale is sufficient to require a variable graphic scale which is obtained by subdividing the latitude borders into minutes and fractions of minutes. The minute of latitude at all latitudes is for practical purposes equal to a nautical mile, hence the subdivided latitude scale is readily applicable to the measurement of correct, distances provided the portion of such latitude scale used for the measurement is at the same mean latitude as the mean latitude of the line whose distance is to be measured. The natural scale given on a coastal or general chart is applicable only over a small portion of the chart and at a stated latitude.

The computed projection is accurately laid down on a copper plate by an engraver and forms the grid upon which the chart is to be built. In cases where the chart is to be engraved on nearly the same scale as that of the survey sheets it is the practice to photograph the survey sheets to the scale of the copper plate projection. Photo prints are then prepared in the drafting room to indicate to the engraver just what is to be engraved, the soundings to be selected, the names to appear on the engraving, their position, style, and size, topographic details and symbols to be employed, and the delineation of fathom curves.

In cases where the chart to be engraved is on a considerably smaller scale than that of the survey sheets it is the practice to first have the outlines engraved on the plate from photographic reductions of the survey sheets, the engraver generalizing to such extent as the difference in scale requires. A dry proof, *i. e.*, a proof printed on dry paper and thus practically free from distortion, is then printed from the plate and furnished to the draftsman who draws thereon the hydrography, topography, and lettering, making

such selection of soundings and details as the difference in scale between the survey sheets and the chart requires. The drawing thus made is then furnished to the engraver for completing the engraving on the plate.

As previously stated, the surveys upon which a large portion of the charts published by the Hydrographic Office are based must of necessity be of foreign origin. These foreign surveys reach the Hydrographic Office in the form of charts which have been constructed and published by the nation which made the surveys. The style and standards used by the several nations differ to a considerable extent and are not in conformity with the practices of the Hydrographic Office. For instance, Japanese charts have soundings expressed in meters while the Hydrographic Office uses the fathom as the unit of depth, except on certain large scale harbor charts where the soundings are expressed in feet. The depth curves given on Japanese charts are meter curves while those shown on Hydrographic Office charts are fathom curves. Hence, in constructing Hydrographic Office charts from Japanese surveys it is necessary to convert the soundings from meters to fathoms or feet, as the case may be, and to delineate proper fathom curves. The same is true in dealing with the surveys of several other nations which are actively engaged in carrying on hydrographic surveys.

GEOGRAPHIC POSITIONS.

In the construction of general charts and coastal charts many individual surveys must be combined. Frequently surveys made by several different nations, at different times and with various degrees of accuracy and completeness, must be combined in compiling a single chart. In making such a compilation conflicting data and descrepancies in geographic positions are usually found. It therefore becomes necessary to thoroughly investigate the relative reliability of the various surveys being used and make such adjustments as will produce a resulting chart with the least probable error.

On an ideal chart the geographic positions of all topographic and hydrographic features should be correct. In other words each feature should be shown in its correct latitude and longitude.

If this were possible to attain then the true geographic relation between topographic and hydrographic features throughout the world could be presented on a series or combination of such ideal charts.

It is possible to determine numerous astronomic latitudes and longitudes along the coasts and within the interior of a continent, connect such positions with systems of precise triangulation and adjust mathematically to determine the most probable position of the whole continent with reference to the equator and an initial meridian. Such determinations and adjustments have been made for North America by the United States Coast and Geodetic Survey in the establishment of the North American datum.

There are vast areas of the earth's surface where systematic determinations of geographic positions have never been attempted and where for various reason no such determinations are likely to be undertaken. These areas, however, must be charted, using such local determinations of geographic positions as may be available and adjusting topographic and hydrographic features between such positions in such a manner as to produce as accurate a chart as possible.

CHART PRINTING.

After the engraving of a chart on copper is completed copies may be printed direct from the engraved plate. This is accomplished on a plate printing press which consists of a bed plate on which the engraved plate rests and which passes between two cylinders under pressure. The cylinder which presses against the face of the engraved plate is covered with a heavy wool blanket.

In the operation of plate printing the engraving is first filled in with ink. The surplus ink is wiped from the face of the engraving with cheese cloth and then the surface is given a final wiping and polishing with bare hands. The paper which has previously been thoroughly wet is laid over the face of the engraving, and run through the press under pressure. The wool blanket causes the wet paper to be pressed into the lines of the engraving thus picking up the ink and giving a clear cut and black impression of the engraving.

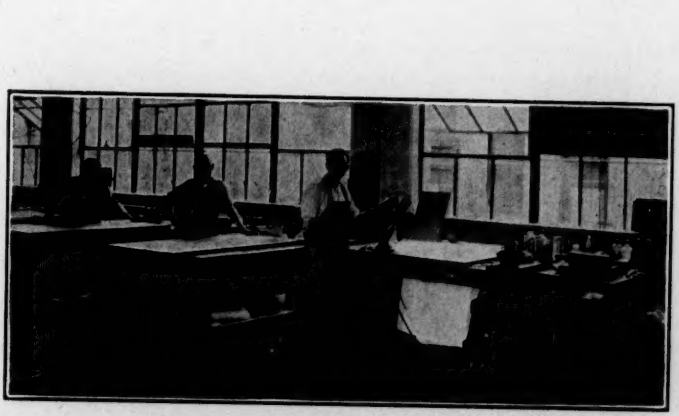
This method of chart printing is slow and laborious. Seventy-five to one hundred copies per day is the average output for one press. The resulting charts are considerably distorted. The paper being wet when printed shrinks unevenly in drying, the shrinkage in one dimension of the sheet frequently being double that in the other direction.

PHOTO-LITHOGRAPHIC PROCESS.

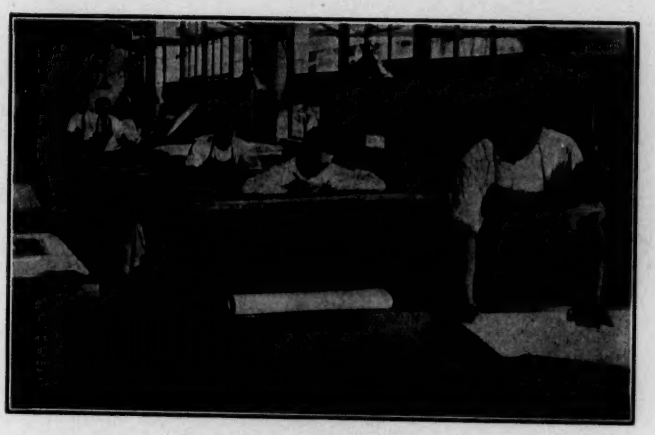
In 1912 the Hydrographic Office adopted the lithographic offset process for printing its nautical charts. This process allows of printing on dry paper at the rate of from one to four thousand impressions per hour, thus producing a more accurate chart at considerably less cost.

At the time the lithographic process was adopted the Office had over sixteen hundred engraved chart plates. These have gradually been transferred to zinc plates for lithographic printing until there remains only about three hundred plates which have to be plate printed by hand. These remaining plates are mostly worn to such an extent by constant printing that they are unfit for transfer to zinc, and must be replaced sooner or later by new engravings. However, the engraved copper plate is still considered the best base or original from which to produce charts and the engraving of such plates from which to pull impressions for photolithographic reproduction is continued.

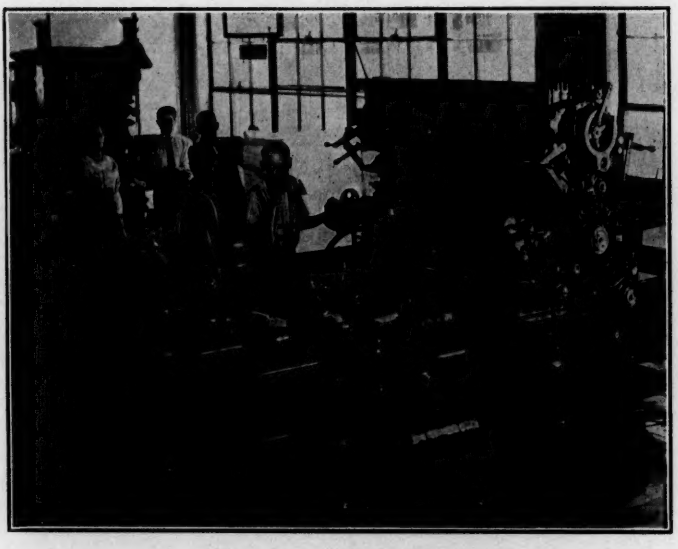
In order to produce photo lithographed charts free from distortion a special process of pulling an undistorted impression from an engraved plate has been developed. For this purpose an unsensitized photographic paper is used. The paper is first coated on one side with a specially prepared paste and allowed to dry. Just before an impression is to be taken this paper is laid between two moistened sheets of paper until only sufficient moisture is absorbed to render the paste tacky. The paper is then laid face down on the engraved plate which has previously been inked in the same manner as for ordinary plate printing. A sheet of blotting paper is then laid on in contact with the paste coated surface and the whole run through a plate printing press. The pressure thus applied causes the photographic paper to adhere firmly to the blotting paper and a sharp and clear impression of the engraving



LITHOGRAPHIC TRANSFERRING.



LITHOGRAPHIC DRAFTING ROOM.



LITHOGRAPHIC OFFSET PRESS.

is obtained on the face of the photographic paper. The next day, after the ink is dry, another sheet of paste coated paper is applied under pressure to the back of the blotting paper thus forming a sort of cardboard. Impressions taken in this manner are found to be free from distortion and their dimensions are practically the same as the engraving itself. Such impressions are known as mounted originals.

When editions of charts were printed entirely by plate printing from the engraved plates it was necessary to engrave every feature appearing on the charts. There are certain features such as compass roses with magnetic variation, variation curves, light sectors, and restricted areas, which are subject to frequent change. The erasing and changing of such features on the engraving is exceedingly costly and injurious to the engraved plates. In the present practice of engraving charts for photo-lithographic reproduction such changing features are omitted from the engraving and supplied either by drafting on the mounted original or by lithographic transfer. Also, all standard notes, such as light notes, conversion tables, publication notes, and seals are not engraved on the copper plates, but are supplied by lithographic transfer.

PHOTO-LITHOGRAPHIC REPRODUCTION OF CHARTS.

After a mounted original has been pulled from an engraved plate, and the necessary features added by a draftsman, it is ready to be photographed. The camera used for the purpose is a wet plate process camera, adapted for negatives 25 inches X 30 inches in size. As most charts are larger than the size of a single negative it is necessary to photograph a chart in sections which are later accurately assembled in photo-printing to a lithographic zinc plate. For this purpose match marks are placed on the mounted original in such positions that two or more marks are common to adjacent negatives. Great care is necessary in keeping the camera and copy board in proper adjustment to prevent distortions and inaccuracies in the negatives.

After the negatives have been made they are placed in the hands of a negative cutter who paints out all defects such as pin holes and dust marks in the background and recuts any weak or defec-

tive work. Paper photo-prints are then make from the negatives and thoroughly revised before photo-printing on zinc.

The zinc plates used in lithographic offset printing are approximately twelve-thousandths of an inch thick and have one surface grained. The grained surface is obtained by placing the plate in a graining machine which consists of a flat tray in which the plate rests. The plate is firmly clamped in place and covered with porcelain marbles about one inch in diameter. The tray is mounted on eccentric bearings and geared to a motor in such a manner that an oscillating motion is imparted to it causing the marbles to roll continuously over the surface of the plate. A very fine quartz sand and water are sprinkled over the marbles which while in motion cause the sand to cut into the surface of the zinc plate thus producing the grained surface. About forty-five minutes is required for the operation of graining a plate.

A zinc plate is prepared for photo-printing by coating the grained surface with a sensitizing solution composed of albumen and bi-chromate of ammonia. The solution is poured onto the surface of the plate which is clamped in a whirler and revolved so as to spread and dry the sensitized coating uniformly. The plate is then ready for printing.

In photo-printing the chart negatives on the plate, a vacuum printing frame is used. The negatives are laid on the surface of the plate and printed one at a time by exposing before arc lamps, the surface not covered by a negative being blocked out and protected with opaque paper. After the first negative has been printed the match marks are developed and the next adjacent negative laid on the plate with its match mark fitted perfectly over those printed from the first plate, and so on with as many negatives as are required to complete the chart.

The effect of light on the sensitized coating is to render the portions exposed to light insoluble in water. After the photo-printing is completed the plate is covered with a dye solution and placed under a water tap. The water dissolves and washes off all of the coating on the plate which has not been exposed to light leaving the design of the chart. The lines of the design consist of an insoluble skin adhering firmly to the surface of the plate and visible in the color of the dye used.

The zinc plate is now ready to receive lithographic transfers, of compasses, notes, seal, etc. A lithographic transfer is an impression or print made on a paste coated paper with a greasy ink specially adapted to the purpose. The print is made on the paste coated side of the transfer paper so that there is a layer of paste between ink and paper thus preventing the ink from penetrating the paper. Standard plates containing compass roses and other features common to charts are used to pull the transfers from.

The operation of transferring consists of inking up a standard plate, laying a piece of transfer paper over the design, and running it through a press under pressure. The transfer paper is then stripped from the surface of the standard plate and the design appears in greasy ink on the paste coated surface of the paper. The paper transfer is then ready to lay down on the chart plate. The desired positions of the work to be transferred are carefully laid out on the chart plate and the transfers laid-on face down in the positions required. Great care must be taken to properly orient transfers of compass roses. The plate is next run through a press under pressure which causes the transfers to adhere firmly to the plate. Water is then applied to the back of the transfers which soaks through the paper and dissolves the paste coating. The paper is then readily removed leaving the transferred design in greasy ink, clear and sharp on the surface of the chart plate.

After the transfer of compass roses, marginal notes, etc., has been completed the zinc plate goes to a lithographic draftsman who draws the border lines of the chart and touches up the design along the joining lines of the several negatives. This is done with a greasy substance known as lithographic tusche which is applied to the plate in the same manner as drawing ink on a paper drawing.

The plate is now ready for etching and preparing for the press. The etching is done with a gum solution containing phosphoric acid which is applied to the surface of the plate with a brush. After being etched the plate is rolled up in ink with a hand roller, the surface of the plate being kept moist with water to prevent the ink from adhering other than to the lines of the design. The surface of the plate is then fanned dry and bezine and asphaltum applied. Which removes the ink and leaves in its place an asphal-

tum coating on the lines of the design. The plate is now ready for the press.

The principle of lithographic printing is a separation between grease and water. During printing the lithographic plate passes alternately under dampening rollers which keep the surface of the plate moist and ink rollers which supply ink to the lines of the design. The lithographic offset press contains three cylinders. The plate is wrapped around and clamped to the first cylinder, a rubber blanket is stretched around and fastened to the second cylinder, and the paper on which the printing is done passes around the third cylinder. While printing the cylinders are in contact. The rubber blanket takes the impression from the zinc plate and imparts it to the paper. The design on the plate is direct or forward. It becomes reversed on the rubber blanket, and direct again on the paper.

In the early days when charts were hand printed direct from the engraved copper plates the land areas were covered with a stipple tint consisting of parallel lines of dots spaced equidistant apart. The ruling of this tint was exceedingly laborious and costly and very difficult to replace neatly when corrections within the tinted area were necessary.

In the lithographing of charts a flat color tint is used over the land areas. This is printed in the same manner as the base design using ink of any desired color. The plate for a land tint is prepared by blue printing the chart design on a zinc plate using the same negatives as were used for the base plate. A ferro-prussiate solution is used for this purpose instead of the bi-chromate sensitizing solution used in photo-printing the base plate. The print thus obtained will not work up and print in the lithographic press, but serves as a guide to the lithographic draftsman who fills in the entire land area with lithographic tusche. The land plate is then etched and prepared for the press in the same manner as the base plate. Similarly plates are prepared for tinting water areas blue out to the three or five fathom curves.

On Hydrographic Office charts printed by lithography the color scheme adopted is black for the base, buff tint for the land, blue tint for water areas out to the three or five fathom curves, green tint over areas dry at low water and orange circles at lights. The

green color over dry at low water areas is obtained by an overlay of the blue and buff tints.

Before the printing of each color on an edition of a chart a proof is printed and thoroughly revised to insure accuracy in the finished charts.

UP-TO-DATE CHARTS.

After a nautical chart is completed, printed, and placed on issue, the work in connection with that particular chart is by no means at an end. Every chart issued must be correct to the date of issue and all available information must be considered to insure its correctness. For that reason the charts are printed in small editions and the supply in stock is kept corrected by hand.

As soon as information is received which necessitates corrections to a chart which can not be readily made by hand arrangements for printing a new edition are made. If the corrections required are not too extensive they are readily made by a draftsman on the lithographic zinc plates. Frequently, however, it becomes necessary to prepare an entirely new zinc plate in which event the copper plate engraving on which the chart is based is brought up-to-date, a new mounted original printed and reproduced by photo-lithography in exactly the same manner as a new chart.

In addition to the regular nautical charts the Hydrographic Office publishes various special charts such as great circle sailing charts, magnetic charts, time zone chart, cable charts, track charts, and pilot charts.

The great circle sailing charts of the several oceans are constructed on gnomonic projections. A straight line connecting any two points on these charts is an arc of a great circle. A great circle course may be laid down on a Mercator chart by first drawing a straight line between the two ends of the desired course on the great circle sailing chart and then reading off its intersections with the meridians and parallels of the gnomonic projection and plotting the same on the Mercator chart. A curved line passing through the plotted points on the Mercator chart will be the great circle track which can be navigated by making an appropriate change of course at fixed intervals.

The magnetic charts consist of three charts based on an outline chart of the world on the Mercator projection and having curves of equal magnetic variation, curves of equal magnetic inclination or dip, and curves of magnetic horizontal intensity, over printed in colors. These charts are compiled and new editions printed every five years. Those now on issue have curves for the year 1925 and give the annual rates of change in the values of the magnetic variation. The next edition will be for the year 1930.

The magnetic variation chart is used in determining the value of variation given on the compass roses of the regular nautical charts. An effort is made to change the variation values on the regular charts every five years, but due to pressure of other work many charts are allowed to go with variation unchanged for a longer period if the application of the annual rate of change to the values given will produce correct results.

The problem of furnishing the Navy and Merchant Marine with adequate nautical charts is one of ever increasing difficulty. The increase in draft of the modern vessels necessitates much greater detail in the delineation of hydrography. Shoals and banks over which light draft vessels could navigate freely now become dangers and a menace. The increase in hydrographic surveying activities throughout the world adds greatly to the burden of the chart makers in providing new charts and revising old ones. Extensive improvements in port and harbor facilities render obsolete charts of little value.

The Hydrographic Office is dependent to a large extent upon the users of nautical charts in obtaining information as to changes which are constantly occuring throughout the world. Constructive criticism of charts is helpful, but active cooperation on the part of navigators in furnishing information is absolutely necessary in order to provide the best and most complete charts.

SCREW PROPELLERS—FALLACIES AND FACTS.

By Rear Admiral C. W. Dyson, U. S. N., Retired.

THEORIES SHOULD BE DEVELOPED TO FIT THE FACTS, NOT THE FACTS DISTORTED TO FIT THE THEORIES.

For many years past the screw propeller and its peculiarities of performances have been subjects of the greatest interest both to engineers, operating and designing, and to mathematicians. Elaborate theories have been developed regarding the phenomenas of performance, while the practical man has been exercised to obtain a set of thumb rules by which the propeller for any vessel could be easily and quickly computed.

The mystery surrounding the subject produces that peculiar charm for the human mind which mysteries always seem to do, so that it is not surprising that in any gathering of marine engineers the subject of the screw propeller is certain to arise sooner or later.

Suppose such a gathering of engineers to occur and after discussing various subjects, that of screw propellers is brought up. Mr. B states that in his experience he has found that a certain form of blade section uniformly gives the best results. Several other members of the gathering disagree with him. Each advocating some other form, but no two agreeing. Finally in an attempt to end the discussion X makes the following statement:

INFLUENCE OF FORM OF BLADE SECTION ON THE PERFORMANCE OF THE PROPELLER.

"Figure 1 shows a number of blade sections of propellers taken close to the hub. These sections, with slight variations, cover practically the entire field of blade section form. The letter E indicates the entering edge of the section and L the leaving one.

"Section A is that most commonly used, and has its back formed as a circular arc.

"Section B has its greatest thickness one-third of the width from the leading edge on the theory that by so locating it, edge resistance of the blade through the water is decreased.

"Section C has the thickness of the blade equally divided on each side of the nominal pitch surface, the front and back of the section being formed with circular arcs, while

"Section D also has the thickness divided but the greatest thickness is only one-third the width of the blade from the leading edge.

"Section E has both the leading and the following edges thrown back from the nominal pitch face. Sometimes only the leading edge is so thrown while in others only the following edges.

"Section F is that commonly known as 'Aerofoil,' and is used for propellers designed by the 'Circulation Theory.'

"Section G is a modification of Section C. The greater part of the thickness is here carried in the back of the blade.

"Section H is a section where the straight portion of the back is at such an angle that it will run at true slip while the face runs at apparent slip.

"All of these sections are satisfactory when of such fineness that neither the entering nor the leaving edges cause eddying. Section H is particularly bad when the blade is very thick and causes, in such cases, an abnormal resistance to turning."

Mr. S here spoke up and stated, "You have spoken of Section F. I wish to state that this section greatly reduces resistance to turning and also greatly increases efficiency of propulsion."

X resuming, states that, "provided blade sections are sufficiently fine to prevent eddying at the entering and leaving edges and all the thickness is back of the nominal pitch surface, the only effect that change in section form produces is to change the angle between the actual and nominal pitch surfaces to a very slight degree, so slight in fact as to make no appreciable difference in revolutions at any given speed and power." Turning to Mr. S, he then continues,—"this I will show you, together with the fallacy of your claim concerning gain in efficiency, when I discuss the performances of a certain vessel later, as disclosed by model tank trials."

Mr. C here injects himself into the conversation with the query, "What has been the experience of you gentlemen with forms of projected areas? I adhere always to the standard oval form, and usually with very good results." Others immediately joined in,

some advocating one form of projection, some another, and one dilating lengthily upon the value of what he called the "tug-boat" type of blade.

X again enters into the discussion and states that after listening to the different participants, he can only come to one conclusion, which is that none of them have any conception as to the effect of blade projected form on propeller performance; that each seems to consider that after having determined on pitch, diameter and projected area of the propeller, any blade projected form appealing to the artistic taste of the designer may be arbitrarily adopted. He then goes on to state his own views as follows:

STANDARD PROJECTED AREA FORMS AND VARIATIONS.

"One of the greatest difficulties met with in seeking for the proper solution of the propeller problem has been produced by the great variations in blade forms and blade sections encountered.

"The difficulties encountered in solving any problem increase rapidly as the number of variables in the problem increase. It is therefore desirable to reduce these variables to as low a number as possible.

"In my own work, in order to do this, I have adopted as a standard for radial distribution of projected area the forms shown by (1) and (2), Figure 2, varying the projected areas by increasing or decreasing the lengths of circular arcs at any radius proportionately to the ratios between the old and new projected areas. Thus, (1) is the half projection of one blade of a 3-bladed propeller whose projected area ratio, outside of the .2 radius, is .25, while (2) is for a projected area ratio of .333⅓. The arc lengths at any radius for (1) and (2) are as 3 to 4, the same as the ratios of the projected areas.

"The standard form of blade section adopted is that shown by A, Figure 1, the ratio of thickness to width decreasing as the blade tips are approached and also as the projected area ratio is increased.

"The actual blades may be thrown back, forward, trailed back from or advanced towards the direction of rotation. The tips may be thrown forward or backward. In fact any change may be made in the actual appearance of the projection so long as the standard radial distribution of projected area for that particular projected area ratio is preserved.

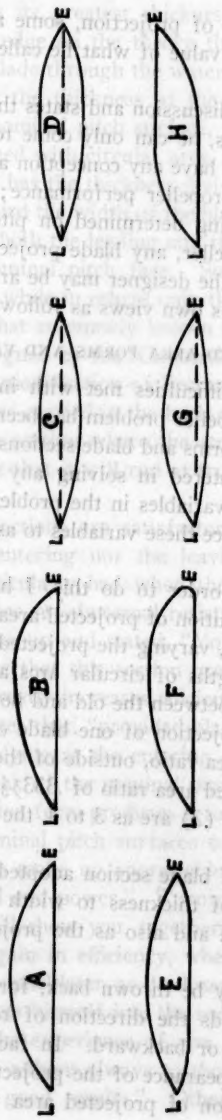


Fig. 1.

VARIATIONS OF FORMS.

"Only these variations of form from the standard are tolerated. The first of them being shown by (1) and (2) of Figure 2, where the blades tips have been cut off at the radius arc $a - a$.

"Such blades are not so powerful as the standard blade of full diameter and will turn up at higher revolutions, thus calling

Full diameter = D
Reduced diameter = D_1
I.H.P. of full blade = I.H.P.
I.H.P. of red. blade = I.H.P.$_1$

$$\text{I.H.P.}_1 = \text{I.H.P.} \times \left(\frac{D_1}{D}\right)^2$$

Power loss factor, full blade = K

$$\text{Power loss factor, red. blade} = K \left(\frac{D}{D_1}\right)^3$$

Effective horsepower, full blade = e.h.p.
Effective horsepower, red. blade = e.h.p.$_1$

Then when K = 1

$$\text{e.h.p.}_1 = (\text{e.h.p.}) \times \left(\frac{D_1}{D}\right)^2."$$

"When K is greater than 1, e.h.p.$_1$ will be decreased and the propulsive efficiency decreased.

"Speed of ship for the full and the reduced blades being the same, then

Revolutions full blade = R
Revolutions red. blade = R_1

$R_1 = R \times \sqrt[4]{\frac{D}{D_1}}$ when out of cavitation

$R_1 = R \times \sqrt{\frac{D}{D_1}}$ when in cavitation.

"This method of diameter reduction should never be used where the power loss factor K exceeds unity.

"The second variation of form is shown by (3) and (4), Figure 2. This modified form of blade is radically different in

its action from that of (1) and (2) with tips cut off to $a - a$. Blades similar to (3) and (4) are of the same power as the full blade of (1) and (2) but for equal powers and speeds of ship the revolutions are higher, thus:

I.H.P. = Indicated horsepower of (1) or (2)

$I.H.P._1$ = Indicated horsepower of (3) or (4)

D = Diameter of (1) or (2)

D_1 = Diameter of (3) or (4)

$I.H.P._1$ = I.H.P.

e.h.p. = Effective horsepower delivered by (1) or (2)

$e.h.p._1$ = Effective horsepower delivered by (3) or (4)

$e.h.p._1$ = e.h.p.

R = Revolutions of (1) or (2) with I.H.P.

R_1 = Revolutions of (3) or (4) with $I.H.P._1$ = I.H.P.

$$R_1 = R \times \sqrt{\frac{D}{D_1}}$$

$\frac{P.A.}{D.A.}$ = Projected Area ratio of (1) or (2)

$\left(\frac{P.A.}{D.A.}\right)_1$ = Projected Area ratio of (3) or (4)

$\left(\frac{P.A.}{D.A.}\right)_1 = \frac{P.A.}{D.A.}\left(\frac{D}{D_1}\right)^2$, the total

projected areas of (1) and (3) and of (2) and (4) being equal.

"This form of projection is desirable to use where the revolutions desired are low for the power to be absorbed, for the allowable diameter of the propeller and for the wake of the vessel. It is particularly desirable where a standard form of projection at these revolutions and power would put the propeller in or on the verge of entering cavitation due to excess thrusts. It is not desirable to use it where the standard form is on the verge of entering or has already entered cavitation due to excess tip speeds.

"One of you referred to the 'tug-boat' type of blade. There is no such type. The conditions for a tug-boat when towing are those which indicate the use of reduced forms such as (3) and (4)

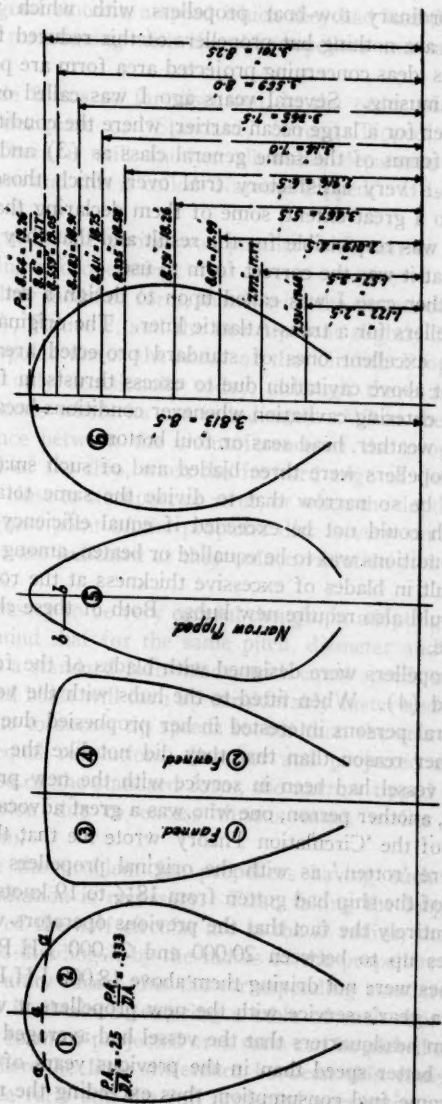


FIGURE 2.

and the ordinary tow-boat propellers with which we are all acquainted are nothing but propellers of this reduced form.

"People's ideas concerning projected area form are peculiar and, at times, amusing. Several years ago I was called on to design the propeller for a large ocean carrier, where the conditions forced me to use forms of the same general class as (3) and (4). The vessel had a very satisfactory trial over which those on board enthused to a great extent, some of them declaring that the blade form used was responsible for the result and that they had always believed that it was the correct form to use.

"In another case I was called upon to design a set of replacement propellers for a trans-Atlantic liner. The original propellers were most excellent ones of standard projected area form but plotted just above cavitation due to excess thrusts in fair weather conditions, entering cavitation whenever conditions became adverse due to bad weather, head seas or foul bottom.

"The propellers were three bladed and of such small projected area as to be so narrow that to divide the same total projected area, which could not be exceeded if equal efficiency under fair weather conditions was to be equalled or beaten, among four blades would result in blades of excessive thickness at the root. Such a change would also require new hubs. Both of these changes were undesirable.

"The propellers were designed with blades of the form typified by (3) and (4). When fitted to the hubs with the vessel in drydock, several persons interested in her prophesied due failure and for no other reason than that they did not like the blade form. After the vessel had been in service with the new propellers for some time, another person, one who was a great advocate of design by means of the 'Circulation Theory' wrote me that the new propellers were 'rotten,' as with the original propellers the original operators of the ship had gotten from 18½ to 19 knots out of her, omitting entirely the fact that the previous operators were driving the engines up to between 20,000 and 21,000 I.H.P., while the present ones were not driving them above 18,000 I.H.P.

"After a year's service with the new propellers it was reported to me from headquarters that the vessel had averaged three-tenths of a knot better speed than in the previous years of her service, with the same fuel consumption, thus exceeding the mail-contract

speed by a good amount, a speed which they had had the greatest difficulty in maintaining with the old propellers and same power of engines.

"The blade forms so far discussed, in the order of their discussion, I designate as 'Standard,' 'Broad Tipped' and 'Fan Shaped.' There is still another allowable departure from the standard form and this is called 'Narrow Tipped.' It is shown by (5), Figure 2.

"This form at one period was popular with marine engineers, both designing and operating.

"In a slightly modified form, that is with the tip cut off at $a - a$, (1) and (2), Figure 2, and the new tip bent slightly forward, it was very popular under the name of the 'Griffith Propeller.' It is met with only occasionally today, although there are a large number of cases where its use is indicated. When there is very little clearance between the circumference of the propeller disc and the hull of the ship, when the forward edge of the propeller blade passes close to sternpost or strut, or when the depth of water over the highest point of the propeller disc is very small, narrow tipped blades are indicated as they reduce vibration and the power loss factor K.

"It is necessary to be very careful though, in adopting them and to bear in mind that for the same pitch, diameter and projected area ratio as a standard form blade, with equal speeds of ship the narrow tipped blade will turn up considerably faster and will enter cavitation due to excess tip speeds sooner than the standard form blade; for equal revolutions, diameter and projected area being equal to those of the standard form, the narrow tipped blade will enter cavitation due to excess thrusts much earlier than the standard form.

"To solve the problem for propellers having such blades, a double computation is necessary. First, taking the length of the chord $b - b$ of the half circular arc at the .9 radius, shown on 5, Figure 2, and dividing it by the radius of the propeller, the chord length, with unity radius, for the corresponding standard form at the .9 radius is obtained; entering a table of such chord lengths, the standard form of projected area ratio can be picked out.

"Should there be little difference between this equivalent tip projected area ratio and the full projected area ratio, in the case

of a 3-bladed propeller, ¾-ths of the total for 4-bladed and 3/2-d's in the case of two blades, then calling the

$$\text{Equivalent projection area ratio at tip} = \left(\frac{\text{P.A.}}{\text{D.A.}}\right)_T, \quad (a)$$

$$\left.\begin{array}{l}\text{4 blades; } ¾ \times \text{Full} \\ \text{3 blades; } 1 \times \text{Full} \\ \text{2 blades; } {}^3/_2 \times \text{Full}\end{array}\right\} \text{ Projected area ratio} = \left(\frac{\text{P.A.}}{\text{D.A.}}\right)_F, \quad (b)$$

the mean projected area ratio (Standard) for the propeller will be

$$\left\{2 \times \left(\frac{\text{P.A.}}{\text{D.A.}}\right)_T + \left(\frac{\text{P.A.}}{\text{D.A.}}\right)_F\right\} \div 3.$$

"The problem can now be solved directly by using this mean projected area ratio.

"Should, however, there be a considerable difference between the equivalent standard projected area ratio of the tip and the projected area ratio as found by (*b*), it becomes necessary to solve the problem for power and revolutions by using the $\left(\frac{\text{P.A.}}{\text{D.A.}}\right)_T$, and then solving it again using $\left(\frac{\text{P.A.}}{\text{D.A.}}\right)_F$.

"Calling I.H.P._T and R_T the power and revolutions derived by using (a), and I.H.P._F and R_F those by using (*b*), the actual power and revolutions for the propeller will be

$$\text{Indicated horsepower} = (2 \times \text{I.H.P.}_T + \text{I.H.P.}_F) \div 3.$$
$$\text{Revolutions} = (2 \times R_T + R_F) \div 3."$$

"In these results attention is called to the much greater influence the tip has upon propeller performance than the remainder of the blade."

Mr. W, who up to this time has remained a silent listener to the discussion, enters with a statement to the effect that he has had experience with propellers of variable pitch whose performances he could not equal with any propeller designed by himself and inquires of Mr. X concerning the latter's opinion on the subject of variable pitches, pitches varying radially, axially and the combination of both. He also desires to know the effect on the pitch of the blades at different radial distances out from the center produced by twisting the blades in their hub seatings.

RADIALLY AND AXIALLY VARYING PITCHES.

"In reply to your questions, Mr. W, should conditions under which each and all blades of a propeller operate be constant throughout a complete revolution, there being a constant variation in velocity of flow of the water through the blades from the hub out to the tips of the blades and from the leading to the following edges, and the velocities of flow of the water entering the propeller at the forward (leading) edges of the blades, together with the rate of the increase in velocity, be known, the proper pitches at each radial distance out from the hub and at the following edge of the blades could be computed with some degree of accuracy, and a slight increase in propulsive efficiency would probably be realized.

"Such intimate knowledge as to the velocities and variations in velocity are not known; neither are the conditions of pressure and velocity constant throughout a revolution. These are considerably higher in the lower semicircle of blade travel than they are in the upper, and the direction of flow in relation to the direction of revolution of the propeller is in the left hand semicircle of revolution just the opposite to what it is in the right hand one. The problem is further complicated by the great variation in after body lines of ships hull, every variation in which will produce more or less variation of the conditions under which the propeller operates.

"The best that could possibly be done would be to design the pitch at each radial distance out from the hub to fit the mean velocity of flow at that radius throughout a revolution. We have not, however, even sufficient knowledge to do this, but are really forced by lack of such knowledge, to design the pitch for the mean of these radial conditions. Anyone claiming differently can have but little idea of the vital effect of the character of the afterbody lines upon the flow of water to the propeller and upon the propulsive efficiency which can be obtained.

TWISTING BLADES IN HUB SEATINGS TO CHANGE PITCH.

"In reply to your question concerning the effect on pitch produced by twisting the blades in their hub seatings, I can not do better than to call your attention to Figure 3.

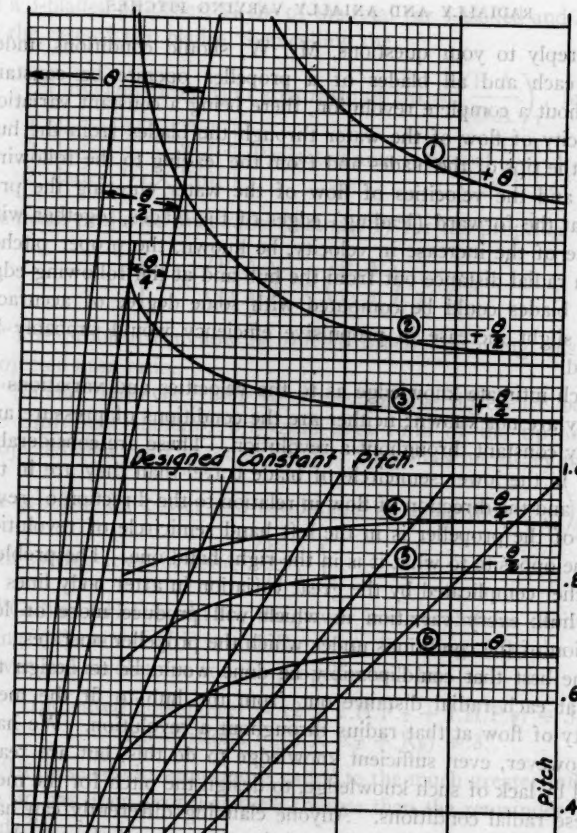

θ
θ/2
θ/4
+θ
+θ/2
+θ/2
Designed Constant Pitch.
-θ/4
-θ/2
-θ
1.0
.8
.6
.4
.2
0
Pitch
Radius of Prop.
π D = Circ. of Prop.
Rad. of hub
π Diam. hub
.2
.4
.6
.8
1.0

Fig. 3

"This figure has for its abscissas values or πD for the fractional values of D shown on the scale of abscissas. The hub diameter, D_H, is taken as .2 D. The constant pitch is P. Now, twist the blade through an angle Θ, so that the angle at the hub becomes 90 degrees. The pitch at the hub has then become infinite while that at the tip has only increased to 1.6 P. The curve of pitches resulting from this is shown by (1).

"Should the angle of increase be reduced to $\frac{1}{2}\Theta$, the curve of resulting pitches is shown by (2), while for $\frac{1}{4}\Theta$, it has dropped to (3).

"Instead of twisting the blades to increase the pitch, twist them in the other direction through $\frac{1}{4}\Theta$. The resultant pitch curve is shown by 4; through $\frac{1}{2}\Theta$, by (5); through Θ, by (6); through a sufficient angle to bring the pitch at the tip to zero, the resultant curve is shown at (7), the blade pitch then be radially expanding from the hub out to about 2/5 the radius, after which it becomes radially decreasing."

Leaving this subject a Mr. A spoke up to the effect that he would like to know the consensus of opinion concerning cavitation. He said that he based his propeller work on a certain theory and did not admit the existence of the phenomenon erroneously called cavitation. He did recognize a point in performances which he called the "burble" (?) point and that this point was produced by the interference of the hull with the flow of water to the propeller. He further stated it to be his opinion that if the propeller could be located at a great distance abaft the hull, no "burble" point, or if others chose to call it so, cavitation, would ever be encountered.

All the others present disagreed with Mr. A, the majority of them claiming that cavitation undoubtedly existed, but at only high speeds of ship and high tip speeds of propellers. Only a few were inclined to recognize the existence of cavitation at low speeds of vessel, low engine powers and low revolutions.

Several of those present then brought up the question of revolutions, as to whether they depended upon speed of ship, power of engine being absorbed by the propeller, or both. Some advocating one point of view, some another, and still others another.

Mr. M then broke in with: "I would like to know the general opinion of the best number of blades to use for the propellers of

twin screw ships: I am heartily in favor of four-bladed ones, basing my preference on the following experience:

"Several years ago the firm with which I am connected built a twin screw vessel and fitted her with two three-bladed propellers. These propellers, in service, were very unsatisfactory, the vessel having very heavy vibrations. We removed these propellers and fitted new ones having the same diameter, pitch and exactly the same projected area as the original propellers but having this area divided among four blades instead of three. The results were very satisfactory.

"With the same engine power quite a gain in speed was realized and all vibration had disappeared."

Some agreed with Mr. M in his contention. Others advocated the three-bladed propeller, while still others stated that in their opinion there was no preference provided the tips of the blades passed the hull at a sufficient distance to insure that the effect of the beat of the blades passing the hull would be so reduced as not to augment the natural vibration of the hull by timing in with it.

Mr. X again joined in the discussion with, "I am convinced from my experience in propeller work covering a long period of years, that cavitation does exist, not only cavitation at high ship speeds, engine powers and speeds of revolution of the propeller but also at low ones. I will try to explain myself.

CAVITATION.

CAVITATION DUE TO EXCESS TIP SPEEDS.

"Suppose a vessel fitted with an engine and propeller has the engine driven at a constant horsepower, no matter what the revolutions. Further suppose that the resistance of the hull gradually be decreased so that the speed for the constant power increases. The revolutions will increase with the speed at first, but at a slightly less rate, then as the speed further increases, they will increase almost at the same rate as the speed, the propulsive efficiency remaining constant throughout.

"Finally a critical speed will be reached after passing which a rapid fall in propulsive efficiency will occur and the propeller will become erratic in its operation, but the revolutions will again increase with the speed but at a reduced rate from that of speed increase.

"At this critical speed point, while the speed of the ship plus the hydraulic and atmospheric heads are sufficient to keep solid water up against the propeller blade backs, they are not sufficient to give the flow of water through the propeller demanded by the pitch × revolutions, so that the faces of the blades are covered by cavities and froth instead of solid water.

"The phenomenon which occurs upon passing this critical speed is that of cavitation due to excess tip speeds. It is the type of cavitation most generally recognized. Its appearance depends upon the characteristics of the propeller, upon the character of the afterbody lines of the ship and upon the speed of the ship. The question of power being absorbed by the propeller does not enter into the problem.

CAVITATION DUE TO EXCESS THRUSTS.

"Let us now turn in the other direction and, holding the power still constant, gradually increase the resistance. The revolutions will decrease at first, but at a lower rate than that of the speeds but the propulsive efficiency will remain constant.

"Finally, as the speed decreases with increase in resistance, a point will be reached after passing which the efficiency will rapidly decrease while the decrease in speed for a small increase in resistance will become abnormal. The effect upon revolutions will depend upon the type of lines of the ship and upon the wake.

"The phenomena here encountered are those accompanying what I designate as 'cavitation due to excess thrusts.'

"Such cavitation, the existence of which is denied by many engineers and naval architects, depends upon the type of lines, wake and speed of ship, and in addition, upon the gross load being delivered by the propeller. By gross load is meant not only the net effective horsepower required for towing the hull at the given speed, but the augment of this power created by the losses due to the hull form and the position of the propeller relative to the hull. These losses I indicate by the factor K, which is called the power loss factor.

"Cavitation due to excess thrusts can occur at low powers, speeds and revolutions as well as at high ones. Where the propeller performance plots so as to indicate the existence of both types of cavitation existing at the same time, the cavitation due to

excess tip speeds must be disregarded and the propeller treated as being in cavitation due to excess thrusts only, as this latter produces the greater loss of the two, and the two can not exist at the same time.

"At the instant that cavitation due to excess thrusts occurs the demand of the propeller for water supply, as expressed by pitch × revolutions, has become so great that the speed of the ship ahead plus the velocity of flow of the water due to the hydraulic and atmospheric heads can not supply it to the back of the propeller, while the water in the rear of the propeller presses solidly up against the blade faces. The backs of the blades are covered by cavities and foam while the faces of them bear hard back against the solid water following them.

"To explain the difference between the two types of cavitation I shall use a homely illustration: Suppose a man is crossing a swamp using his feet to thrust in the mud while with his hands he grasps a rope, stretching from one side of the swamp to the other, and pulls. Both hands and feet are doing their share of the work necessary to get him over. Suddenly his feet strike in quick sand against which he can get no foothold. His feet immediately fail to help him and all the work of moving him ahead is put upon his hands. His feet (the propeller blade faces) have failed him and his salvation depends entirely upon his hands (the propeller blade backs).

"Suppose, however, that he meets with no quicksands but that the going gets heavier and heavier until finally the strain he is putting on the rope becomes so great as to break it. All the work of propulsion is thrown upon the feet which begin to sink much deeper in the mud with every step and whose rearward slip becomes much increased. His hands (the blade backs) have become useless while the entire load has been thrown upon his feet (the blade faces).

"The first condition approximates closely to that of cavitation due to excess tip speeds, while the second, cavitation due to excess thrusts.

"In answering your queries concerning revolutions and effect of number of blades I shall be forced to return to this subject of cavitation.

RELATIONS EXISTING BETWEEN REVOLUTIONS, POWER OF ENGINES AND SPEED OF SHIP.

"When a propeller is out of cavitation the revolutions depend upon both power and speed, the influence of the latter, however, upon the apparent slip being considerably greater than that of the former.

"The equation for apparent slip, the propeller not being in cavitation is

$$s = (S \times 10^{Z_S}) \div \times 10^{Z_E}$$

or, written as a logarithmic equation,

$$\text{Log. } s = \text{Log. } S + Z_S \mp Z_E$$

s = apparent slip at speed v.

S = Basic apparent slip = measure of wake.

Z_S = Speed factor = 2.869 Log. $(V \div v) - .0788$.

V = Basic Speed = Critical Speed beyond which propeller cavitates due to excess tip speeds.

Z_p = Net load factor.

K = Power loss factor.

$Z_E = Z_p \mp \text{Log. } K$ = Gross load factor.

Z_p = 1.0414 Log. (E.H.P. ÷ e.h.p.) where e.h.p. is less than E.H.P.

Z_p = 1.0414 Log. (e.h.p. ÷ E.H.P.) where e.h.p. is greater than E.H.P.

e.h.p. = Actual tow-rope horsepower required for the speed v.

E.H.P. = Basic tow-rope horsepower.

I.H.P. = Basic indicated horsepower.

S.H.P. = Basic shaft horsepower = .92 I.H.P.

I.H.P._d = Actual indicated horsepower for the speed v.

S.H.P._d = Actual shaft horsepower for the speed v.

$$\left.\begin{matrix}\text{I.H.P.}_d \\ \text{S.H.P.}_d\end{matrix}\right\} = \left\{\begin{matrix}\text{I.H.P.} \\ \text{S.H.P.}\end{matrix}\right\}\left\{\begin{matrix}\div \\ \times\end{matrix}\right\} 10^{Z_E}, \text{ or}$$

$$\text{Log.} \left\{\begin{matrix}\text{I.H.P.}_d \\ \text{S.H.P.}_d\end{matrix}\right\} = \text{Log.} \left\{\begin{matrix}\text{I.H.P.} \\ \text{S.H.P.}\end{matrix}\right\}\left\{\mp\right\} Z_E.$$

"Now, Mr. A, did I understand you to ask the meaning of the word *basic* as I use it?

"I will try to explain as clearly as possible.

"In making my investigations of the phenomena encountered in the field of propeller performances I chose a series of vessels whose block coefficients ranged from about .6 down to about .4. These vessels were all very fine lined, both as to water lines and buttocks. They were all fitted with two propellers, three blades to each propeller, and with projected area ratios measured outside a circle of .2 radius, varying from .315 to .6. The projected areas were of the same radial distribution, the lengths of the arcs at each radius being proportional to the projected area ratios. This distribution of areas was similar to those shown in the blades (1) and (2) of Figure 2. The pitch of each propeller was constant and uniform while all the propellers had blade sections similar to A, Figure 1. The driving faces of the propellers were machined to pitch and backs finished to template by hand. All blades were highly polished.

"The after bodies of the hulls were all long as compared with the draught and the lines were very fine, the propellers being located well abaft the hull fullness and with good clearance between the propeller discs and the hulls. The propellers were all three-bladed.

"The vessels were carefully run over the measured mile at the specified displacements corresponding to those at which the models had been towed in the tank. The vessels were just out of dock with clean freshly painted bottoms. The observers for taking data of speeds, powers and revolutions, were all highly skilled. The vessels, in order to eliminate the effect of tidal currents on the speed data, were run twice in one direction and once in the opposite for the lower points while for the highest performance points three in one and two in the opposite direction were run, the directions in which the runs were made alternating for each point.

"The propulsive efficiencies of these propellers were all high; they were as high as, and in some cases considerably higher than, had been realized with any equal projected area ratio propeller for which I had the records. I therefore assumed that these efficiencies were the maximum for each projected area ratio tested, when driving hulls of maximum hull efficiency, that is hulls for

which the power loss factor K is unity, K being the measure of the power loss due to thrust deduction and to hull interference with the flow of water to the propellers."

"From the resultant plotted curves of speed—Indicated Horsepower, Speed—Revolutions, Speed—Effective Horsepower, I obtained for each speed and projected area ratio:

$$I.T._D = \frac{I.H.P._d \times 33000}{P \times R \times 36\,\pi\,D^2} = \text{Indicated Thrust per square inch of disc area.}$$

$S.T._D = .92\ I.T._D$ = Shaft Thrust per square inch of disc area.

$T.S. = \pi \times D \times R$ = Tip Speed.

e.h.p. = Effective Horsepower for each speed.

$I.H.P._d$ = Indicated Horsepower for each speed.

$S.H.P._d = 92\ I.H.P._d$ = Shaft horsepower for each speed.

$$P.C. = \frac{e.h.p.}{I.H.P._d} = \text{Propulsive Coefficient on Indicated Horsepower.}$$

$$P.C._S = \frac{e.h.p.}{S.H.P._d} = \text{Propulsive Coefficient on Shaft Horsepower.}$$

"Plotting the values of T.S and P.C. for each projected area ratio on the corresponding values of $I.T._D$, a series of curves were obtained.

"The curves of P.C. plotted one above the other, in the order of the projected area ratios, that for the minimum ratio showing the highest propulsive efficiencies. All of them, however, after reaching a certain value of $I.T._D$, depending upon the projected area ratio, began to decrease in value, at first slowly, but soon dropping quite rapidly; the lower projected area ratio curves of P.C. showing this phenomenon earlier than the greater ones.

"These points where the rapidity of drop began were at the conditions where the propellers were just entering cavitation due to excess tip speeds. They all lay on a fair enveloping curve, probably hyperbolic in character.

"Taking these points on the enveloping curve, and their corresponding points of T.S. and $I.T._D$ for each projected area ratio, and plotting them on a chart having values of projected area ratios, P.A. ÷ D.A., as abscissas, the values of P.C. were found to lie on a curve for which the equations are

From O to P.A. ÷ D.A. = .2

$$P.C. = .9074\ (P.A. \div D.A.)^{.125\left(1+\frac{P.A.}{D.A.}\right)}$$

From P.A. ÷ D.A. = .2 to P.A. ÷ D.A. = .55

$$P.C. = \frac{11.85}{25.84}\left(\frac{(P.A. \div D.A.) - .0875}{(P.A. \div D.A.)^{1.7}}\right)$$

From P.A. ÷ D.A. = .56 to P.A. ÷ D.A. = .75

$$P.C. = \frac{18.75}{28.54}\left(\frac{(P.A. \div D.A.) - .265}{(P.A. \div D.A.)^{1.7}}\right)$$

"After passing P.A. ÷ D.A. = .75, blade interference occurs and the efficiency rapidly decreases.

"The values of $I.T._D$ were found to lie on a curve whose equation is

$$I.T._D = 28.54\left(\frac{P.A.}{D.A.}\right)^{1.7}$$

while that for tip speeds is

$$T.S. = 38148\left(\frac{P.A.}{D.A.}\right)^{.8365+2\left(\frac{P.A.}{D.A.}\right)}$$

The only part of these curves which interest us in the study of hydraulic screw propellers extends from

$$\left(\frac{P.A.}{D.A.} = .2 \text{ to } \frac{P.A.}{D.A.} = .65.\right)$$

The propulsive coefficient curve between

$$\left(\frac{P.A.}{D.A.} = O \text{ and } \frac{P.A.}{D.A.} = .2\right)$$

is of interest, however, as it is applicable to aeroplane and dirigible propellers as it stands.

"The conditions given by the above curves are called by me, basic conditions of the propeller. There is still another basic condition, that of the basic apparent slip, which I will take up later on.

TO FIND THE BASIC I.H.P., S.H.P. AND E.H.P. OF THREE AND FOUR BLADED PROPELLERS.

"The assertion is often made that a three-bladed propeller is more efficient than a four-bladed one, or the opposite position may be taken. A three-bladed propeller having the same diameter,

pitch and total projected area ratio as a four-bladed one, should, when both propellers are working, under similar conditions and out of cavitation, be slightly more efficient than the four on account of the lesser number of blade edges around which eddying may take place; also, the blades of the three being wider and thinner than those of the four, the tendency towards eddying at the blade edges is reduced. The difference in the efficiencies, if any, is so slight that it is covered in the errors of data for calculation purposes.

"The blades of the four-bladed propeller will function as those of a three-bladed having only three-fourths of the projected area of the four, but the basic propulsive coefficient will be that of a three-bladed propeller having a projected area equal to the total projected area of the four-bladed propeller.

"To illustrate, I will take two propellers, one of three blades, the other of four, having the same diameter, pitch and projected area ratio, using a total projected area ratio of .333, which is the standard form of blade projection shown by (2), Figure 2, for a three-bladed propeller of that projected area ratio. Three-fourths of .333 gives .25, which is the standard form shown at (1), Figure 2, and which will be the form for each blade of the four-bladed propeller.

"Then for

Three-bladed		Four-bladed	
Diameter	17′	Diameter	17′
Pitch	14′	Pitch	14′
Total P.A. ÷ D.A.	.333	Total P.A. ÷ D.A.	.333
P.A. ÷ D.A.	.333	P.A. ÷ D.A.	.25
T.S. for P.A. ÷ D.A.	7320	T.S. for P.A. ÷ D.A.	5981
$I.T._D$ for P.A. ÷ D.A.	4.45	$I.T._D$ for P.A. ÷ D.A.	2.73
$I.H.P. = \frac{T.S. \times P \times D \times I.T._D}{\pi \times 291.8}$	8471	$I.H.P. = \frac{T.S. \times P \times D \times I.T._D}{\pi \times 252.41}$	4901
S.H.P. = .92 I.H.P.	7780	S.H.P. = .92 I.H.P.	4509
P.C. for Total P.A. ÷ D.A.	.662	P.C. for total P.A. ÷ D.A.	.662
E.H.P. = I.H.P. × P.C.	5599	E.H.P. = I.H.P. × P.C.	3244

There is still missing from the data and the computed quantities obtained from the data, the basic slip and the resultant basic (critical) speeds, neither of which appear in the above tabulated data and computations.

TO OBTAIN THE BASIC SLIP AND BASIC SPEED.

"The wake of a ship is dependent upon the fullness of afterbody lines of the vessel, upon the fullness of midship section, upon the ratio of beam to height of stream line body above the base line or to ratio of beam to draught in some cases.

"The basic apparent slip for any vessel is dependent upon the wake of the vessel and upon the condition of the water in the wake.

"The measure of fullness of the afterbody is determined graphically and the result obtained is called the slip block coefficient, S.B.C. The Power Loss Block Coefficient, K.B.C., is also obtained graphically from the same chart, but it must be remembered that the values of the loss factor K given by the chart of K are for the orthodox forms of hulls which are usually encountered. Cases are met where with an apparently fine lined hull K becomes K^2 while in other cases the actual values of K will be much reduced below those given by the chart. I have just had some cases where the K.B.C. was as high as .86 but the actual K was unity, due to the great length of afterbody and hollowness of afterbody lines of the ships and possibly to the depth of water in which the ship operated.

"Having obtained the S.B.C. of the ship graphically, the basic apparent slip can be obtained from the following equation:

$$A(1 - S)^{\alpha} = \frac{2\,\text{L.A.B.}}{H^1}, \text{ where}$$

L.A.B. = Length of afterbody of the ship.

H^1 = Height of stream line body above base.

$$A = 4.1\left(\frac{10\,(1 - \text{S.B.C.})}{3}\right)^{\frac{1}{\text{Log. 10 S.B.C.} - .34}} + 12.8$$

but Log. 10 S.B.C. − .34 must never be less than .23

$$\alpha = .1112(A)^{.9582 - .000806A}.$$

"The values of S are, however, plotted graphically for values of S.B.C. and 2 L.A.B. ÷ H^1, and can be picked off without calculation.

"I will now return to the three and four bladed propellers. Let us suppose that the basic slip of the vessel which is being driven be 13.25 per cent, then

S..... .1325	S..... .1325
1-S..... .8675	1-S..... .8675
$V = \frac{T.S. \times P \times (1 - S)}{\pi D \times 101.33}$ 16.43	$V = \frac{T.S. \times P \times (1 - S)}{\pi D \times 101.33}$ 13.42

"To summarize briefly the three-bladed propeller and four-bladed propeller will be on the verge of cavitating due to excess tip speeds when absorbing

I.H.P. 8471.....4901
S.H.P. 7780.....4509

and delivering a gross

E.H.P. 5599.....3244

with the ship traveling at a speed of

V.....16.43 knots.....13.42 knots

through the water (*not over the bottom*). The powers are, however, absolutely independent of the speeds, depending only upon the characteristics of the propellers.

PERFORMANCE OF THE PROPELLER IN CONDITIONS NOT BASIC.

"Calling the actual speed of the ship through the water v, and the gross effective horsepower required $e.h.p._g$, this being the effective horsepower for the speed v plus losses due to the hull, it is very seldom that we find the speed fraction, $\frac{v}{V}$, equal to unity and at the same time the load fraction $\frac{e.h.p._g}{E.H.P.}$ also equaling unity. It becomes necessary to provide means of computing powers and revolutions at other than basic conditions.

COMPUTATIONS FOR POWER AT CONDITIONS NOT BASIC.

"Let $I.H.P._d$ = Actual Power.
$e.h.p._g$ = Gross Effective Power.
v = Actual Speed.
$\frac{e.h.p._g}{E.H.P.}$ = Load fraction.
$\frac{v}{V}$ = Speed fraction,

then

$$\frac{\text{I.H.P.}}{\text{I.H.P.}_d} = \left(\frac{V}{v}\right)^w \text{ or}$$

$$\text{Log. I.H.P.} - \text{Log. I.H.P.}_d = w \text{ Log.} \left(\frac{V}{v}\right)".$$

"For all propellers, no matter what their dimensions, so long as the sections are sufficiently fine to prevent eddying at the edges, the surfaces fairly smooth and the projected areas distributed proportionately to (1) and (2), Figure 2, so long as $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ is held constant the value of

$w \text{ Log.} \left(\frac{V}{v}\right)$ will remain constant; from $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 0$,

up through all possible values these values of $w \text{ Log.} \left(\frac{V}{v}\right)$, which I call Z_g, plot in a curve whose equation is

$$Z_g = 1.0414 \text{ Log.} \left(\frac{\text{E.H.P.}}{\text{e.h.p.}_g}\right)$$

when e.h.p._g is less than E.H.P., and

$$Z_g = 1.0414 \text{ Log.} \left(\frac{\text{e.h.p.}_g}{\text{E.H.P.}}\right)$$

when e.h.p._g is greater than E.H.P.

When the power loss factor K is unity, the power factor is called Z_p, when K is greater than unity,

$$Z_g = Z_p - \text{Log. K},$$

and in some cases

$$Z_g = Z_p - 2 \text{ Log. K}."$$

"Thus having the basic I.H.P. or S.H.P. of the propeller, the power for any other load condition equals

$\text{I.H.P.}_d = \text{I.H.P.} \div 10^{Z_g}$ where e.h.p._g is less than E.H.P.,

and

$\text{I.H.P.}_d = \text{I.H.P.} \times 10^{Z_g}$ where e.h.p._g is greater than E.H.P.

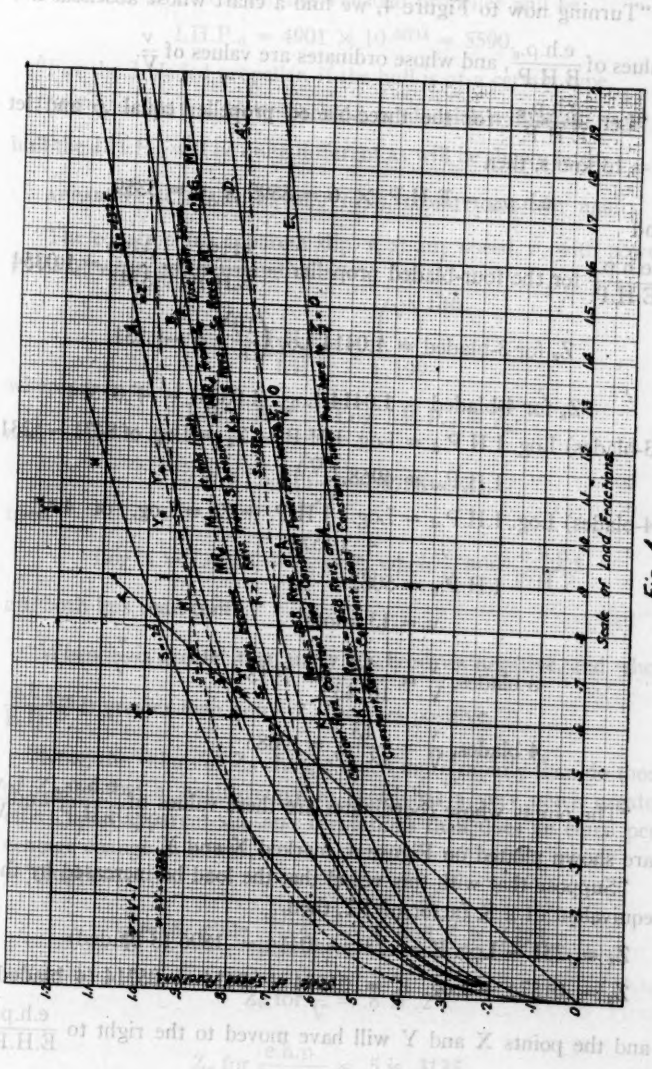


Fig. 4

"Turning now to Figure 4, we find a chart whose abscissas are values of $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$, and whose ordinates are values of $\frac{v}{V}$.

"Let $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ for the three-bladed propeller equal .6 and let v = 13 knots, then

$$\text{e.h.p.}_g = \text{E.H.P.} \times .6 = 5599 \times .6 = 3359$$

and

$$\frac{\text{e.h.p.}_g}{\text{E.H.P.}} \text{ for the four-bladed propeller} = \frac{3359}{\text{E.H.P.}} = \frac{3359}{3244} = 1.0354$$

$$Z_g \text{ for 3-bladed} = 1.0414 \text{ Log.} \left(\frac{10}{6}\right) = - .231$$

$$Z_g \text{ for 4-bladed} = 1.0414 \text{ Log. } (1.0354) = + .01574$$

$$\text{(3-blades) Log. I.H.P.}_d = \text{Log. I.H.P.} - Z_g = \text{Log. of } 8471 - .231$$

$$\text{I.H.P.}_d = 4968.$$

$$\text{(4-blades) Log. I.H.P.}_d = \text{Log. I.H.P.} + Z_g = \text{Log. of } 4901 + .01574$$

$$\text{I.H.P.}_d = 5082.$$

$$v = 13 \text{ knots}$$

$$\text{3 blades, } \frac{v}{V} = \frac{13}{16.43} = .7913$$

$$\text{4 blades, } \frac{v}{V} = \frac{13}{13.42} = .9685$$

"The points whose coordinates are these values of $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ and $\frac{v}{V}$ are shown plotted on Figure 4, marked X and Y.

"Suppose that with this vessel that the load be increased by the equivalent of a K factor of 1.1, then

$$Z_g = .231 - \text{Log. K} = .231 - .0414 = .1896 \text{ (3 bladed)}$$

$$Z_g = .01574 + \text{Log. K} = .01574 + .0414 = .05714 \text{ (4 bladed)}$$

and the points X and Y will have moved to the right to $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ = .655 for the 3 and $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ = 1.135 for the 4-bladed propeller.

"The new I. H. $P._d$ for the 4-bladed propeller will be

$$I.H.P._d = 4901 \times 10^{.05714} = 5590.$$

"For the 3-bladed propeller, if the hull is of a certain type, $I.H.P._d = 8741 \div 10^{.1896} = 5465$, while with a different type of hull the $I.H.P._d$ will be quite different as will be shown later.

APPARENT SLIP AND REVOLUTIONS, CONDITIONS NOT BASIC.

"The equation for apparent slip, s, at any speed, v, with power, $I.H.P._d$, when not in cavitation, is

$$s = S \frac{I.H.P._d}{I.H.P.} \times \frac{\left(\frac{V}{v}\right)^{2.869}}{1.199}$$

where

$$\text{Log. I.H.P.} - \text{Log. I.H.P.}_d = Z_g$$

and

$$2.869 \text{ Log.} \left(\frac{V}{v}\right) - .0788 = Z_s,$$

then

$$\text{Log. } s = \text{Log. } S + Z_s - Z_g \text{ when } \frac{v}{V} \text{ and } \frac{e.h.p._s}{E.H.P.}$$

are both less than unity.

"When $\frac{v}{V}$ is greater than unity, Z_s becomes negative, and when $\frac{e.h.p._s}{E.H.P.}$ is greater than unity, Z_g becomes positive.

"Now, the values of Z_s change much more rapidly than do those of Z_g, so that a small change in speed has a very much greater effect upon apparent slip and revolutions than does an equal percentage change in power.

"Suppose

$$\frac{v}{V} = .8, \text{ while } \frac{e.h.p.}{E.H.P.} = .5, \text{ and } S = .1325$$

$$Z_s \text{ for } \frac{v}{V} = .8 \text{ is } .2$$

$$Z_g \text{ for } \frac{e.h.p.}{E.H.P.} = .5 \text{ is } .3135,$$

therefore

$$\text{Log. } s = 9.12222 + .2 - .3135 = 9.00872$$
$$s = .102$$

"Let the speed be increased ten per cent, the power remaining constant, then

$$\frac{v}{V} = .88 \text{ and } Z_s = .08$$

$$\text{Log. } s = 9.12222 + .08 - .3135 = 8.88872$$
$$s = .0774.$$

"The revolutions under the two conditions will be

$$\text{for } \frac{v}{V} = .8 \ldots\ldots \frac{v \times 101.33}{P \times .892}$$

and

$$\text{for } \frac{v}{V} = .88 \ldots\ldots \frac{1.1\,v \times 101.33}{P \times .9226}$$

"By the increase in speed, the revolutions will have become increased in the ratio

$$\frac{1.1\,v \times 101.33}{P \times .9226} \times \frac{P \times .892}{v \times 101.33} = \frac{.9812}{.9226} = 1.064-.$$

"Suppose now, that the speed remains v where $\frac{v}{V} = .8$, but that the gross effective horsepower for this speed be increased ten per cent, then

$$\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .55 \text{ and } Z_g = .27$$

$$\text{Log. } s = 9.12222 + .2 - .27 = 9.05222$$
$$s = .1128$$

"The ratio of the revolutions will then be

$$\frac{v \times 101.33}{P \times .8872} \times \frac{P \times .892}{v \times 101.33} = \frac{.892}{.8872} = 1.0054$$

"This condition is limited, however, to the range of speeds between that speed at which the propeller with the given load enters cavitation due to excess thrusts and a speed where $\frac{v}{V} = .9385$.

"At this value of $\frac{v}{V}$, $Z_s = O$, and the curve of Z_s breaks, passing rapidly from $Z_s = O$ at $\frac{v}{V} = .9385$ to $Z_s = -.0266$ at $\frac{v}{V} = 1.02$ where the propeller is fully in cavitation due to excess tip speeds.

"Here the speed increase is

$$1.02 \div .9385 = 1.087 -$$

For $\frac{v}{V} = .9385$ and $\frac{e.h.p._s}{E.H.P.} = .5$, $S = .1325$

Log. $s = 9.12222 + O - .3135 = 8.80872$

$s = .06438$

For $\frac{v}{V} = 1.02$, $Z_s = -.0266$, $\frac{e.h.p._s}{E.H.P.} = .5$, $Z_s = -.3135$

Log. $s = 9.12222 - .0266 - .3135 = 8.78212$

$s = .06055.$

"The ratio of the revolutions for the two speeds will be

$$\frac{1.087\ v \times 101.33}{P \times .93945} \times \frac{P \times .93562}{v \times 101.33} = \frac{1.087 \times .93562}{.93945},$$

where we see that with constant power the revolutions in this range vary almost exactly with the speeds.

"Suppose, however, that $\frac{v}{V}$ remains constant at .9385 but that the load be increased from

$$\frac{e.h.p._s}{E.H.P.} = .5 \text{ to } \frac{e.h.p._s}{E.H.P.} = .5 \times 1.087 = .5435.$$

Then Z_s will become .278 and

Log. $s = 9.12222 + O - .278 = 8.84822$

$s = .07051,$

and the ratio of the revolutions becomes

$$\frac{v \times 101.33}{P \times .92949} \times \frac{P \times .93562}{v \times 101.33} = \frac{.93562}{.92949} = 1.0066.$$

"Here again we see that the speed practically controls the revolutions, and to a considerably greater extent than for values of $\frac{v}{V}$ lower than .9385, the effect of power increases in both ranges being very small.

"Having shown you what occurs in the safe ranges, I will now take you into the unsafe ones, those in which either cavitation due to excess thrusts or cavitation due to excess tip speeds occurs. Again your attention is called to Figure 4. Before taking up the subject, however, I will return to our three and four bladed propellers and determine their apparent slips and revolutions under the original loads.

"The values of $\frac{v}{V}$ were .7913 for the 3-bladed and .9685 for the 4-bladed one, for which

$Z_s = .21$ and $-.01$ respectively, while

$Z_g = -.231$ and $+.01574$.

"Then for the three-bladed propeller,

$$\text{Log. } s = 9.12222 + .21 - .231 = 9.00122.$$
$$s = .1003$$

and for the four-bladed one

$$\text{Log. } s = 9.12222 - .01 + .01574 = 9.12796$$
$$s = .1343.$$

Revolutions for the

$$\text{3-bladed} = R_d = \frac{13 \times 101.33}{14 \times (1 - .1003)} = 104.6.$$

$$\text{4-bladed} = R_d = \frac{13 \times 101.33}{14 \times (1 - .1343)} = 108.7.$$

"The foregoing analyses of these two propellers shows that under the original conditions the three-blade propeller is 2½ per cent more efficient than the four-bladed one and turns up at about 3-4/5 per cent lower revolutions, while at the augmented power, the efficiency of the three-bladed is 2⅓ per cent greater than that of the four.

CAVITATION.

AUGMENTATION OF POWER DUE TO CAVITATION.

CAVITATION DUE TO EXCESS THRUSTS.

"I have already stated that with a vessel having constant power on the propeller, as the resistance is increased the speed and revolutions will decrease until a critical speed is reached after which the efficiency of propulsion falls rapidly and the rate of decrease

in speed increases. This critical speed depends upon the character of the afterbody lines of the ship, upon the basic apparent slip, upon the characteristics of the propeller and upon the gross load upon the propeller.

"Curves of critical speed fractions for values of basic slip from 0 to .25 have been prepared and laid down on a chart of which the abscissas are net, gross and cavitating load fractions.

"On Figure 4, are shown two of these curves of critical speed fractions. The full line curve H is that for a basic slip of .25, the ship hulls having either full afterbody water lines, fine buttocks and short afterbodies or fine afterbody water lines, full buttocks and long afterbodies. The full line curve marked A is for the same types of hulls but with a basic apparent slip of .1325.

"Should the vessels have afterbodies of great length as compared with the height of stream line body, and both water lines and buttocks be fine, the curves of critical speed fractions will move down to H^1 for S = .25, and to A^1 for S = .1325.

"Let us consider the case of a vessel having full afterbody water lines, fine buttocks and a short afterbody, and a basic apparent slip = S = .1325.

"As before call

Basic Indicated horsepower	= I.H.P.
Basic Effective Horsepower	= E.H.P.
Basic Critical Speed	= V
Basic Apparent Slip	= S."

"Let us suppose further, that the propeller is absorbing a constant engine horsepower, $I.H.P._d$, and delivering a gross effective horsepower, $e.h.p._g = .5$, but that the resistance is gradually being increased with the speed of the ship through the water decreasing as the resistance increases.

"Finally the speed has decreased to a point where $\frac{v}{V} = .728$ and the propeller plots on the curve A with $\frac{e.h.p._g}{E.H.P.} = .5$.

"There are now several new quantities enter into the calculations:

v at this point is called v_c.

"The original gross load fraction becomes $\frac{\text{e.h.p.}_c}{\text{E.H.P.}}$, and is called the cavitating load fraction, while the original gross load factor becomes the cavitating load factor, Z_c.

"The gross load fraction at entering cavitation is also called the entering load fraction and its corresponding factor, the entering load factor. They are denominated by $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ and Z_1.

"At the point of entry, $\frac{v}{V} = .728$ and $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .5$

$$\frac{\text{e.h.p.}_c}{\text{E.H.P.}} = \frac{\text{e.h.p.}_g}{\text{E.H.P.}} = \frac{\text{e.h.p.}_1}{\text{E.H.P.}}$$

$$\text{and } Z_c = Z_g = Z_1.$$

"Suppose now, that with the same engine power the resistance further increases so that the speed fraction drops to $\frac{v}{V} = .7$, then $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ will be that load fraction whose ordinate for A is .7, or $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} = .445$, for which the entry load factor $= Z_1 = 1.0414 \text{ Log.} \left(\frac{1}{.445}\right) = .3662$ and

$$Z_g = \frac{Z_c + Z_1}{2} = \frac{.3135 + .3662}{2} = .3398,$$ for which the gross load fraction is

$$\text{Log.} \left(\frac{\text{E.H.P.}}{\text{e.h.p.}_g}\right) = \frac{.3398}{1.0414} \text{ or } .32629$$

$$\text{Log.} \left(\frac{\text{e.h.p.}_g}{\text{E.H.P.}}\right) = 9.67371$$

$$\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .4718$$

"The gross effective horsepower has thus fallen, due to cavitation, from .5 E.H.P., to .4718 E.H.P., while the engine power has remained constant.

"Should the power loss factor K, be unity,

$\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ will equal the net effective load fraction $\frac{\text{e.h.p.}}{\text{E.H.P.}}$, both out of and in cavitation. Should it be greater than unity, say K = 1.1, then $Z_p = Z_g + \text{Log. K} =$

$$.3135 + .0414 = .3549 \text{ out of cavitation}$$

$$= .3398 + .0414 = .3812 \text{ in cavitation, for which}$$

$\frac{\text{e.h.p.}}{\text{E.H.P.}}$ (Net load fraction) = .455 out of

and = .43 in cavitation, so that when not cavitating, the propeller will be delivering .455 E.H.P. as tow-rope horsepower while, when in cavitation to the amount of the calculation, the tow-rope power,

$$\text{e.h.p.} = .43 \text{ E.H.P.}$$

"The equation of the factors (exponents) may be written

$Z_c = 2\ Z_g - Z_1$, from which, knowing any two, the third may be obtained.

"This equation may also be put in the form

$$Z_c = Z_g \pm 1.0414 \text{ Log.} \left(\frac{\text{e.h.p.}_1}{\text{e.h.p.}_g}\right), \text{ the minus sign when } \left(\frac{\text{e.h.p.}_g}{\text{E.H.P.}}\right)$$

is greater than unity, the plus sign when less.

"From this equation it is readily seen that when in cavitation due to excess thrusts the engine power absorbed by a propeller depends not only upon the net and gross effective horsepower being delivered by the propeller but also upon the ratio between the gross effective and that gross effective which would be delivered were the propeller just entering cavitation at the speed v and speed fraction $\frac{v}{V}$, and that so far as this augmentation of power is concerned, the gross effective horsepower e.h.p._1 is the basic effective horsepower of the propeller.

AUGMENTATION OF ENGINE POWER CAUSED BY CAVITATION DUE TO EXCESS TIP SPEEDS.

"Let us now reduce the resistance so that the speed increases until v equals the basic critical speed V, the load fraction and engine power remaining constant, as before.

Then $$\text{I.H.P.}_d = \text{constant}$$

$$\text{e.h.p.}_g = \text{constant}$$

$$\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .5 \text{ and is constant}$$

$$Z_g = .3135$$

"The propeller is just entering cavitation due to excess tip speeds and again Z_c, Z_1, $\frac{\text{e.h.p.}_c}{\text{E.H.P.}}$ and $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ enter into the computations.

"Calling the net effective (tow-rope) horsepower for the new speed e.h.p., and that for the critical speed e.h.p.$_1$, the critical speed being the basic speed V, the equation for the new power factor is

$$Z_c = Z_1 \mp 1.50 \text{ Log.} \left(\frac{\text{e.h.p.}}{\text{e.h.p.}_1}\right) \mp \text{Log. K},$$

which may also be written

$$Z_c = Z_1 \pm 1.44 \left(1.0414 \text{ Log.} \left(\frac{\text{e.h.p.}_1}{\text{e.h.p.}}\right) \mp \text{Log. K}\right.$$

$$\text{or } Z_c = Z_{1g} \pm 1.44 \left(1.0414 \text{ Log.} \left(\frac{\text{e.h.p.}_{1g}}{\text{e.h.p.}_g}\right)\right.$$

where Z_{1g} is the gross load factor for the net load fraction $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ corrected for the power loss factor.

"Again we see that the augment of power due to cavitation, but this time cavitation due to excess tip speed, depends upon the ratio between the entry load and the actual load, both with and without K, for the expression Log. $\left(\frac{\text{e.h.p.}_1}{\text{e.h.p.}}\right)$ can be changed to Log. $\left(\frac{\text{e.h.p.}_{1g}}{\text{e.h.p.}_g}\right)$ and the result will be the same. The entry load has again become the basic load so far as any speeds above V are concerned.

"Suppose we have a propeller plotting at Z, Figure 4, assuming for simplicity, that K equals unity. We see that

$$\frac{\text{e.h.p.}}{\text{E.H.P.}} = \frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1.5$$

while

$$\frac{v}{V} = 1.03, \text{ the basic slip S still being } .1325.$$

"The propeller is apparently in cavitation due to excess tip speeds and also in cavitation due to excess thrusts. It can not be in both, however, and that which produces the greater loss will prevail.

"Assume that the net effective horsepower will vary as the cube of the speeds, then

$$\frac{\text{e.h.p.}_1}{\text{E.H.P.}} : \frac{\text{e.h.p.}}{\text{E.H.P.}} :: \left(\frac{V}{V}\right)^3 : \left(\frac{v}{V}\right)^3$$

$$\frac{\text{e.h.p.}_1}{\text{E.H.P.}} = \frac{\text{e.h.p.}}{\text{E.H.P.}} \times \left(\frac{V}{v}\right)^3 = 1.5 \times \left(\frac{1}{1.03}\right)^3 = 1.372$$

$$\frac{\text{e.h.p.}_1}{\text{e.h.p.}} = .9151$$

Z_1 for $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} = 1.3727$ is $+ .1433$.

Therefore $Z_c = .1433 - 1.50 \text{ Log.} \left(\frac{\text{e.h.p.}_1}{\text{e.h.p.}}\right) =$

$$.1433 - 9.94222 = .2011,$$

and Log. I.H.P.$_d$ = Log. I.H.P. + .2011 (1) for cavitation due to excess tip speeds.

"For cavitation due to excess thrusts $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ for $\frac{v}{V}$ of 1.03 equals 1.435 for which

$$Z_1 = + .21310$$

$$Z_p = Z_g \text{ for } \frac{\text{e.h.p.}}{\text{E.H.P.}} = \frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1.5, \text{ is } + .1834$$

therefore

$$Z_c = 2Z_g + Z_1 = 2 \times .1834 + .21310 = .3668 + .2131 + 5799$$

and Log. I.H.P.$_d$ = Log. I.H.P. + .5799 . . . . . (2).

Dividing equation (2) by equation (1), it is seen that the engine powers required due to the two losses vary as

$$\text{Log. I.H.P.}_{d(2)} - \text{Log. I.H.P.}_{d(1)} = .3788$$

$\text{I.H.P.}_{d(2)} = 2.392\ \text{I.H.P.}_{d(1)}$, and cavitation due to excess thrusts prevails and eliminates the other."

REVOLUTIONS IN CAVITATION.

CAVITATION DUE TO EXCESS TIP SPEEDS.

"When a propeller not in cavitation reaches a speed where $\frac{v}{V} = .9385$, the law governing the values of apparent slip changes. Up to this point the equation for basic slip has been

$$s = S\,\frac{10^{Z_s}}{10^{Z_g}}$$

where Z_S generally is found from the equation

$$Z_S = 2.869\ \text{Log.}\left(\frac{V}{v}\right) - .0788.$$

"At $\frac{v}{V} = .9385$, Z_S becomes zero. As $\frac{v}{V}$ increases, the curve of Z_S passes over from $\frac{v}{V} = .9385$, $Z_S = 0$, in a straight line to $\frac{v}{V} = 1.015$ and $Z_S = -.025$ where it joins the cavitating curve for Z_S, for which the equation is

$Z_S = 3.09\ \text{Log.}\left(\frac{v}{V}\right)$, and the equation for apparent slip still remains

$$s = S\,\frac{10^{Z_s}}{10^{Z_g}}."$$

"To find the relative effects of speed increase and power increase on revolutions in this cavitating range, I will again suppose a speed increase of ten per cent, power constant, and a power increase of ten per cent, speed constant, assuming that the starting speed fraction $\frac{v}{V} = 1.015$, and the starting load fraction $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1.0$, then

Z_s for $\frac{v}{V} = 1.015$ is $-.025$

Z_g for $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1$ is O

Log. s = Log. S. $-$ .025 $-$ O; taking S = .1325

Log. s = 9.12222 $-$ .025 = 9.09722

s = .1251

For $\frac{1.1v}{V} = 1.1515$

$Z_s = -.1893$

Log. s = 9.12222 $-$ .1893 $-$ O = 8.93292

s = .08569

(1) Revolutions at $\frac{v}{V} = 1.015$ are $= \frac{v \times 101.33}{P \times .8749}$

(2) Revolutions at $\frac{v}{V} = 1.1515$ are $= \frac{1.1\,v \times 101.33}{P \times .91431}$

$$\frac{\text{Revolutions (2)}}{\text{Revolutions (1)}} = \frac{1.1 \times .8749}{.91431} = 1.053,$$

or the revolutions vary at a considerable lower rate than the speeds, power remaining constant.

"Now, let

(1) $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1$, then $Z_g = O$

(2) $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1.1$, then $Z_g = +.0431$

(1) Log. s = 9.12222 $-$ .025 $-$ O = 9.09722 : s = .1251

For $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = 1.1$

(2) Log. s = 9.12222 $-$ .025 + .0431 = 9.14032

s = .1382

$$\text{Revolutions (1)} = \frac{v \times 101.33}{P \times .8749}$$

$$\text{Revolutions (2)} = \frac{v \times 101.33}{P \times .8618}$$

$$\frac{\text{Revolutions (2)}}{\text{Revolutions (1)}} = \frac{.8749}{.8618} = 1.015$$

"While the power effect on revolutions has increased above that in the previous conditions studied, it is still far below the effect produced by changes in speed.

REVOLUTIONS IN CAVITATION DUE TO EXCESS THRUSTS.

"When this condition is investigated, matters are not quite so simple. When the propeller enters into cavitation its behavior, so far as revolutions is concerned, depends not only upon the characteristics of the propeller but also on the lines of the afterbody of the hull of the ship.

"As a general rule, when vessels have a basic slip of .1325 or less, upon entering cavitation due to excess thrust the factor Z_S, instead of being derived from the equation

$$Z_S = 2.869 \text{ Log.} \left(\frac{V}{v}\right) - .0788, \text{ must be derived from}$$

$$Z_S = 3.09 \text{ Log.} \left(\frac{V}{v}\right).$$

"When the basic slip is in excess of .1325, the values of Z_S are taken from the equation

$$Z_S = 2.869 \text{ Log.} \left(\frac{V}{v}\right) - .0788 \text{ whether out of or in}$$

cavitation due to excess thrusts."

"There are exceptions to this rule, however, for in cases where the afterbodies have fine water lines and comparatively fine buttocks, but where the ratio $H^1 \div B$ = height of stream line body to beam, is equal to or greater than $H \div B$ = draught to beam, and practically the complete horizontal diameter of the propeller (wing propellers) is unprotected by overhanging wetted submerged surface of the hull, the values of Z_S, both out of and in cavitation, and cavitation of both classes, must be taken from the equation

$$Z_S = 3.09 \text{ Log.} \left(\frac{V}{v}\right) \text{ for all values of basic slip S."}$$

"If the afterbody water lines of such vessel are very fine and the buttocks comparatively full, the value of the power loss factor K also becomes squared, even though this fullness of buttocks be not that of the buttocks themselves but be caused by the fitting of spectacle frames.

"Upon entering cavitation due to excess thrusts the laws governing apparent slip, as expressed by the equation for that quantity, may or may not change, depending entirely upon the character of the lines of the afterbody of the ship and upon its basic slip.

"The equation for apparent slip may remain as

$$s = S\frac{10^{Z_S}}{10^{Z_c}},$$

$$\text{Log. } s = \text{Log. } S + Z_S - Z_c, \text{ which I will call equation (1),}$$

or it may become

$$s_c = S\frac{10^{Z_{Sc}}}{10^{Z_c}},$$

$$\text{Log. } s = \text{Log. } S + Z_{Sc} - Z_c, \text{ equation (2).}$$

"Should the vessel have full buttocks, $H^1 \div B$ less than $H \div B$, or spectacle frames causing an apparent great fullness of buttock, fine water lines, and the propeller horizontal diameters practically uncovered by any wetted overhanging hull surface, the apparent slip equation to use is

"When S is equal to or less than .1325,

$$s = S\frac{10^{Z_S}}{10^{Z_c}}$$

"When S is greater than .1325,

$$s_c = S\frac{10^{Z_{Sc}}}{10^{Z_c}},$$ and the propeller will cavitate at the curve of critical speeds for the basic slip of the vessel, this curve following the same law as curves H and A.

"Should the vessels lines be of the same type as just given, but the value of $H^1 \div B$ and of $H \div B$ be great, $H^1 \div B$ being greater than $H \div B$, the apparent slip will follow the laws as just given but cavitation will be delayed until a curve of critical speed fractions, for the basic slip S, but following the same law as H^1 and A^1, is reached.

"Should, however, the afterbody of the vessel be short, with full water lines and fine buttocks, or should it be quite long, with fine buttocks and fine water lines, the latter fulling out rapidly as the load water line is approached so that from about one-half to the full horizontal diameter of the disc of the propeller is covered by the wetted overhanging skin of the hull, then

"Should S be not greater than .1325,

$$s_c = S\,\frac{10^{Z_{Sc}}}{10^{Z_c}}.$$

"Should S be greater than .1325,

$$s = \frac{10^{Z_S}}{10^{Z_c}}.$$

"The first of these hull types will produce cavitation according to the law governing curves H and A, while the second will follow that of H^1 and A^1.

"The definitions of s and Z_S have already been given. I will now explain those of s_c and Z_{Sc}.

"I have already shown you the meaning of the cavitating load factor, Z_c, and the corresponding cavitation load fraction $\frac{\text{e.h.p.}_c}{\text{E.H.P.}}$. Through the abscissa value of $\frac{\text{e.h.p.}_c}{\text{E.H.P.}}$ draw an ordinate cutting the curve of critical speed fractions for the basic slip involved. The value of $\frac{v}{V}$ at the point of cutting will be $\frac{v_c}{V}$, where v_c is the speed at which the propeller will enter cavitation when delivering a gross load $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = \frac{\text{e.h.p.}_c}{\text{E.H.P.}}$, and the speed factor for this value, $\frac{v_c}{V}$, will be Z_{Sc}.

"Let us now take the case of a vessel having full water lines, fine buttocks and a short afterbody, with a basic slip of .1325. Let it enter cavitation with a gross load fraction of .5, the engine power remaining constant but the resistance increasing and producing a gradual diminution of speed.

"The ordinate at $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .5$, cuts the curve of critical speeds for S = .1325, at $\frac{v}{V} = .735$.

At this point, $\frac{v}{V} = \frac{v_c}{V}$, $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = \frac{\text{e.h.p.}_c}{\text{E.H.P.}}$, $Z_g = Z_c$ and $Z_S = Z_{Sc}$.

"The value of s, as already stated is, at this point

$$s = S\frac{10^{Z_S}}{10^{Z_s}} \text{ and also } s_e = S\frac{10^{Z_{S_e}}}{10^{Z_e}} \text{ and } s = s_e.$$

"As the speed still further decreases, the second of these equations prevails and the revolutions will equal

$$R_d = \frac{v_e \times 101.33}{P \times (1-s_e)},$$

and will remain constant until, with a K value greater than unity, $\frac{v}{V}$ has dropped to the curve B, or with K equals unity, to the curve C. Through these ranges, between A and B or A and C, changes in speed having absolutely no effect upon the revolutions.

"At B and C, depending upon K, a factor M enters into the revolutions, the equation then becoming

$$R_d = M\frac{v_e \times 101.33}{P \times (1-s_e)},$$

M decreasing at a slightly less rate than the speed until the speed has decreased to D for K greater than unity and to E for K equals unity; at these curves M reaches a minimum value of .858. From D and E, as the speed decreases, the revolutions remain constant so long as the power remains constant, until finally we have the vessel tied to the dock and straining ahead on her hawsers.

"Let us now turn to the full buttocked, fine water lined type of ship, also cavitating according to the law for H and A. Suppose the gross load fraction and the speed fraction be the same as for the vessel already discussed.

"In this second case, with S = .1325, while Z_S, as Z_{S_c} in the previous case, must be derived from the equation

$$Z_S = 3.09 \text{ Log. } \left(\frac{V}{v}\right),$$

s will be obtained by the equation

$$s = \frac{10^{Z_S}}{10^{Z_e}},$$

Z_S being the speed factor for $\frac{v}{V}$, v being the actual speed. This condition, with speed and power having the same relative effect upon revolutions as when not in cavitation, continues as the speed de-

creases, power remaining constant, until the curve F, for K greater than unity or G for K equals unity, are reached. At these curves the revolutions obtained by using

$$s = S\frac{10^{z_s}}{10^{z_o}}$$

will again equal those obtained by using

$$s_c = S\frac{10^{z_{s_c}}}{10^{z_c}},$$

as was the case upon just entering cavitation, that is

$$R_d = \frac{v \times 101.33}{P \times (1-s)} = M\frac{v_c \times 101.33}{P \times (1-s_c)}.$$

"From these two curves, s_c, v_c and M must be used, M again reaching its minimum value of .858 at the same curves as before, and the propellers acting as in the previous case until we again have the ship straining on her hawsers at the dock.

"The curves B, C, D, E, F and G as drawn only apply to the curve A where S = .1325. Similar curves for any value of S can be constructed, whether the vessel enters cavitation according to the law for H and A or for H^1 and A^1.

THREE AND FOUR-BLADED PROPELLERS.

THREE-BLADED IN CAVITATION DUE TO EXCESS THRUSTS.

"I will now return to the three and four-bladed propellers of equal pitch, diameter and total projected area ratio.

"Let one ship have the full water lined, fine buttocked, short afterbody, and, as before, let S = .1325. The propeller then enters cavitation due to excess thrusts when $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ with its ordinate $\frac{v}{V}$, plots on the curve A.

"I have already increased the gross load by the equivalent of a K factor of 1.1 over the original, for the same speed of 13 knots. By doing this, the gross load fraction moved from X to X_1, where $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .655$, on Figure 4, and the point plots below the curve A and the three-bladed propeller is therefore in cavitation. The same percentage increase in gross load has moved the load fraction

for the four-bladed propeller from Y to Y^1 but this latter point plots above A and the four-bladed propeller is still well out of cavitation.

"It can be stated as a fact, that having two propellers, one of three blades, the other of four, but having equal pitches, diameters and total projected area ratios, both working under the same conditions of resistance and absorbing the same engine power, that propeller having only three blades will enter cavitation due to excess thrusts earlier than the one of four blades.

AUGMENTATION OF POWER DUE TO CAVITATION.

"The gross load fraction of the 3-bladed propeller is .655, and for this

$$Z_g = -.1914.$$

The value of $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$, where $\frac{v}{V} = .7913$ is .64.

The value of Z_1 for $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} = .64$, is

$$Z_1 = .2019.$$

"Therefore

$$Z_c = 2Z_g - Z_1 = .1814$$

and

$$\frac{\text{e.h.p.}_c}{\text{E.H.P.}} = .6696."$$

"The new value for the engine power will be found from

$$\text{Log. I.H.P.}_d = \text{Log. I.H.P.} - Z_c = 3.92793 - .1814 = 3.74653$$

$$\text{I.H.P.}_d = 5579$$

which very nearly equals the 5590 for the 4-bladed propeller out of cavitation. The 3-bladed propeller is now probably vibrating slightly. Any further decrease in speed or increase in load will bring the power in excess of that required for the four-bladed propeller out of cavitation, and the vibrations will be increased. Mr. M probably encountered a similar case in that which he quoted.

"The value of $\frac{\text{e.h.p.}_e}{\text{E.H.P.}}$ is shown plotted at X on Figure 4. The ordinate through this point cuts A at

$$\frac{v_e}{V} = .81, \text{ hence}$$

$$v_e = 16.43 \times .81 = 13.31$$

$$Z_{se} \text{ for } \frac{v_e}{V} = 3.09 \text{ Log.} \left(\frac{1}{.81}\right) = .2828$$

$$\text{Log. } s_e = \text{Log. } S + Z_{se} - Z_e = 9.1222 + .2828 - .1814 = 9.22362$$

$$s_e = .1674$$

"Therefore

$$R_d = \frac{v_e \times 101.33}{P \times (1 - s_e)} = \frac{13.31 \times 101.33}{14 \times .8326} = 115.7."$$

"The revolutions of the four-bladed screw propeller for the same gross load, e.h.p.$_g$, will be 111.

CAVITATION DUE TO EXCESS TIP SPEEDS.

"Suppose the resistance of the vessel be so decreased that with the same engine power as was obtained with $\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = .6$, a speed whose speed fraction equals .99 for the three-bladed propeller, be obtained. The position of the three-bladed propeller plots at X^{111} on Figure 4. It has approached the verge of cavitation due to excess tip speeds.

"With this new speed, the four-bladed propeller has mounted rapidly on Figure 4, and now plots at Y^{11}, far above $\frac{v}{V} = 1$, and is deeply in cavitation due to excess tip speeds."

Mr. B now speaks up and says, "Mr. X, suppose you were called upon to get the three-bladed one out of cavitation or to design a three-bladed propeller to be out of cavitation due to excess thrusts but not to exceed the revolutions of the original propeller when in cavitation, what would you do?"

Mr. X: "If the blades were adjustable, I would set down the pitch sufficiently to make the new $\frac{v}{V}$ and $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$, as derived from

analysis with the new pitch, bring the propeller to plot above the curve A. In doing this, the positions of the propeller as the pitch is gradually lowered, shift upward very nearly along a line of equal thrusts OT; at the point X^{iv}, the propeller would have diameter —17′, pitch = 13.16 and a projected area ratio of .333, while $I.H.P._d$ would have lowered to 5500 and revolutions to 114.7.

"Suppose we design a series of propellers on a curve through X^{iv}, parallel to A, the points on the series, with constant diameter will be, all with three blades,

D	17′	17′	17′	17′	17′
P	13′.04	13′.2	13′.15	13′.19	13′.26
$\frac{P.A.}{D.A.}$	.393	.358	.335	.315	.298
$e.h.p._g$	3695	3695	3695	3695	3695
$\frac{e.h.p._g}{E.H.P.}$	.5	.6	.7	.8	.9
I.H.P.	5754	5601	5508	5448	5410
R_d	115.7	114.2	114.7	114.8	114.6

"This series can be continued down to a projected area ratio of .25, below which it is not advisable to go, as very little can be gained and the blades begin to thicken up abnormally.

"Let us extend this curve passing through X^{iv}, until it passes through Y^1, as shown by the dotted curve just above A, and from points on this curve to the left of Y^1, as at that point the blade form is already $\frac{P.A.}{D.A.} = .25$, design a series of four-bladed propellers for the same gross load. Then

D	17′	17′	17′	17′	17′
P	13′.46	13′.57	13′.71	13′.69	14′
$\frac{P.A.}{D.A.}$	.326	.304	.286	.271	.25
Total $\frac{P.A.}{D.A.}$	.435	.405	.381	.361	.333
$e.h.p._g$	3695	3695	3695	3695	3695
$\frac{e.h.p._g}{E.H.P.}$	.7	.8	.9	1.0	1.135
$I.H.P._d$	6120	5944	5830	5729	5590
R_d	112.1	111.6	110.9	111	110.4

"For equal projected area ratios and equal leeway above cavitation, it will be seen from these two tables, that the three-bladed

propeller will have lower pitch, higher revolutions and higher propulsive efficiency than will the four-bladed one.

"It may be stated as General Rule I:

"I. When revolutions and diameter of the propeller are low in comparison with the gross load to be delivered by the propeller, the indications point towards a propeller with four-blades, either of standard or fan shape, the latter to be used when the revolutions are lower than can be obtained by using the standard form, in order that the propeller may be kept out of cavitation due to excess thrusts.

"II. Should the speed of the ship and the revolutions of the propeller be high, the indications point to a propeller with three blades should we wish to insure against cavitation due to excess tip speeds.

"Cavitation is one of the phenomena of nature. When conditions arise where the demands of the propeller for water supply to and through the propeller can not be met by the heads, of air and water, and the advance of the ship, cavitation in one form or the other is bound to occur.

"The possibility of its occurrence can not be destroyed by the adoption of freak forms of blades or of blade sections, nor can we cooly wipe it out because our pet theory does not recognize it."

Mr. A now states that he is greatly interested in the Circulation Theory method for the design of propellers and claims that his investigations of the performances of many vessels has assured him that he can safely guarantee an increase in speed of from one to four knots by discarding the old propellers and fitting new ones designed by the use of the Circulation Theory.

Mr. S adds that he has designed several propellers by the use of the Circulation Theory, and by assuming the net effective horsepowers delivered by them in actual service, exceptionally high propulsive efficiencies have been obtained.

Mr. X again takes the floor and says that he is fully aware of Mr. A's wonderful paper results but has yet to hear that any of them have been realized in actual service; in fact, he states, he has heard that of practically all which have been tried, the results have been extremely disappointing. He congratulates Mr. S upon the latter's modesty in assuming such net effective horsepowers as to render the propulsive efficiencies remarkable when there was noth-

ing to prevent him from assuming them of such magnitude as to show the propeller world that the age of miracles was still with us.

Mr. X then proceeds: "The only way to determine whether a propeller designed by the Circulation Theory Method, having the variable pitch and aerofoil blade sections, produces any greater propulsive efficiency than a standard propeller with constant pitch and orthodox circular arc backed blade sections, is to try a vessel first with propellers of one type and then of the other, under similar conditions, the propellers having equal diameters, the true mean pitch of the one being equal to the constant pitch of the other, and having the same number of blades of the same form and projected area ratios.

COMPETITIVE TRIALS OF CIRCULATION THEORY PROPELLERS.

"Fortunately I have at hand such a case. Several years ago a company designing propellers by the Circulation Theory Method entered into a tentative contract with one of the largest ship operating corporations of this country, to provide a design for new propellers for one of their passenger vessels. The carrying out of the contract was to be dependent upon the showing made in the model tank, where with the model of the hull being propelled by a model of the new propeller, a gain of, as I recollect, ten per cent in propulsive efficiency was to be shown over the propulsive efficiency delivered by the model of the original propeller. This gain was to be realized at the ordinary service speed of the vessel, 16½ knots.

"The new propeller plans when submitted showed blades similar to (6), Figure 2, which also shows the pitches, both model and actuals, at several radial distances from the center.

"In preparing these designs the designers undoubtedly had at hand the plans of the original propellers, the performance curves of the ship, both trial and service, and probably the lines of the ship, as I know from personal experience that they always demanded such information. This made quite a different proposition from that facing the designer of the original propellers, who was forced to start with only the lines and net effective horsepower of the ship, the designed power and revolutions of the engines.

"The curve of the radial pitches of the model propeller, together with the method of deriving the mean radial pitch of both the

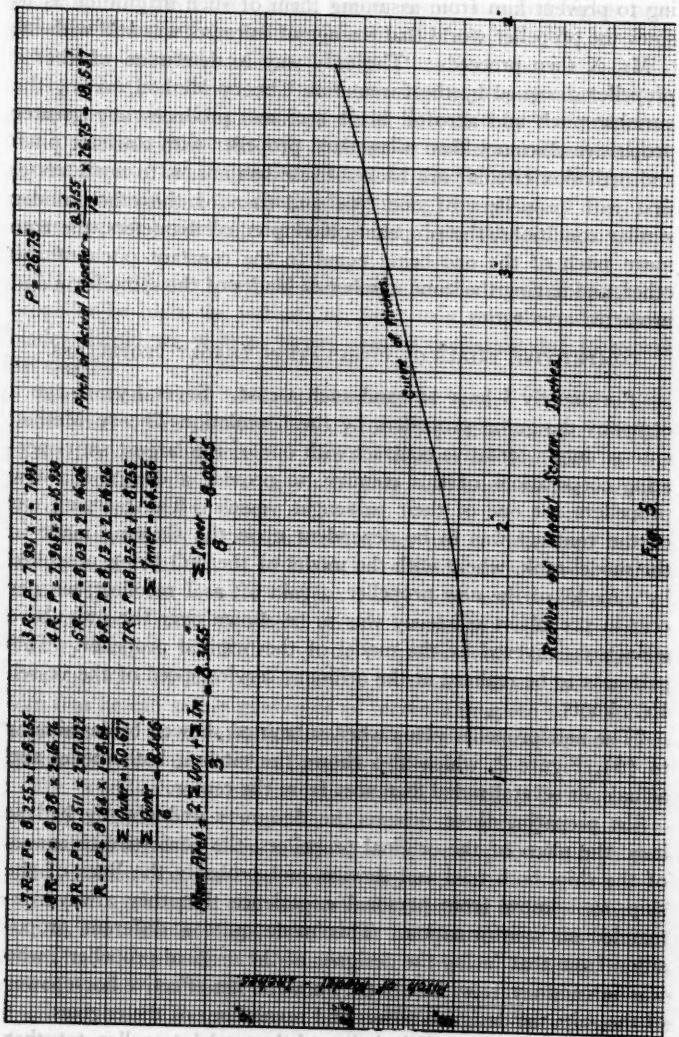

Pitch of Actual Propeller = 8.3455/12 × 26.75 = 18.547
Radius of Model Screw, Inches.
Pitch of Model Screw, Inches.

Fig. 3.

model and the actual propeller, is shown on Figure 5, the ratio of reduction, ρ, being 26.75.

"The blade form of the new propeller at the outer third was equivalent to a projected area form of .3125, while the actual projected area ratio was .317.

"The equivalent standard form was therefore,

$$\frac{2 \times .3125 + .317}{3} = .314.$$

"The mean characteristics of the two propellers, then, were

	Original	Circ. Theory
Blades	3	3
D	17′	17′
P	18′5	18′537
P.A. ÷ D.A.	.315	.314

"In case any change in propulsive efficiency was realized, the owners wished to know whether it was due to the variable pitch, the aerofoil blade sections, or to the two combined; they therefore directed that a third model propeller having the same diameter, constant pitch and projected area ratio of the original propeller but having the aerofoil blade sections of the new one, be constructed and tested in competition with the other two.

"The models being completed, the model hull was run in the model tank at three different displacements, 16013, 20800 and 25863 tons. Each set of propellers driving it at each of these displacements.

"The curves of shaft horsepower and of revolutions for the different speeds obtained with each set of propellers for each displacement and the estimated shaft horsepower and estimated revolution curves, estimated by the Dyson Method which I have explained to you today, are shown on Figures 6, 7, and 8.

"The tank curves are seen to plot remarkably close together, the differences being so slight that they can be attributed to changes in atmospheric, and water conditions between the different runs, and to inherent inaccurracies in the tank instruments and in the personal errors of the observers. They are also seen to plot very close to the estimated curves.

"There probably may be some of you who do not think the changes and errors I have enumerated can produce the differences

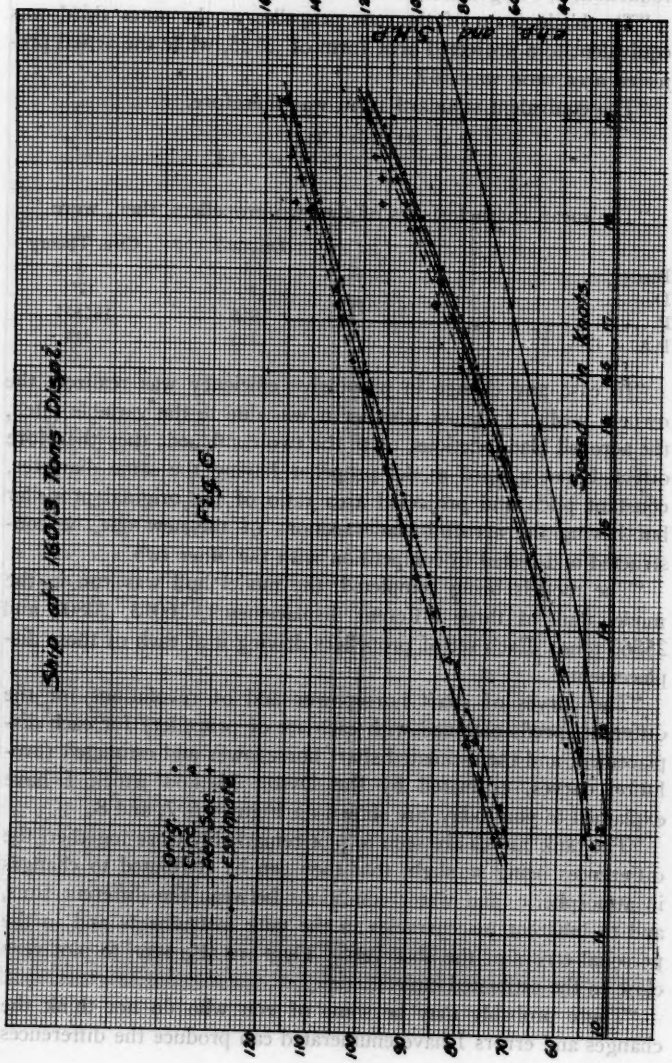


Fig. 6.
Ship of 18013 Tons Displt.
Speed in Knots.
I.H.P.
Orgt. Circ. Propr.
Estimate

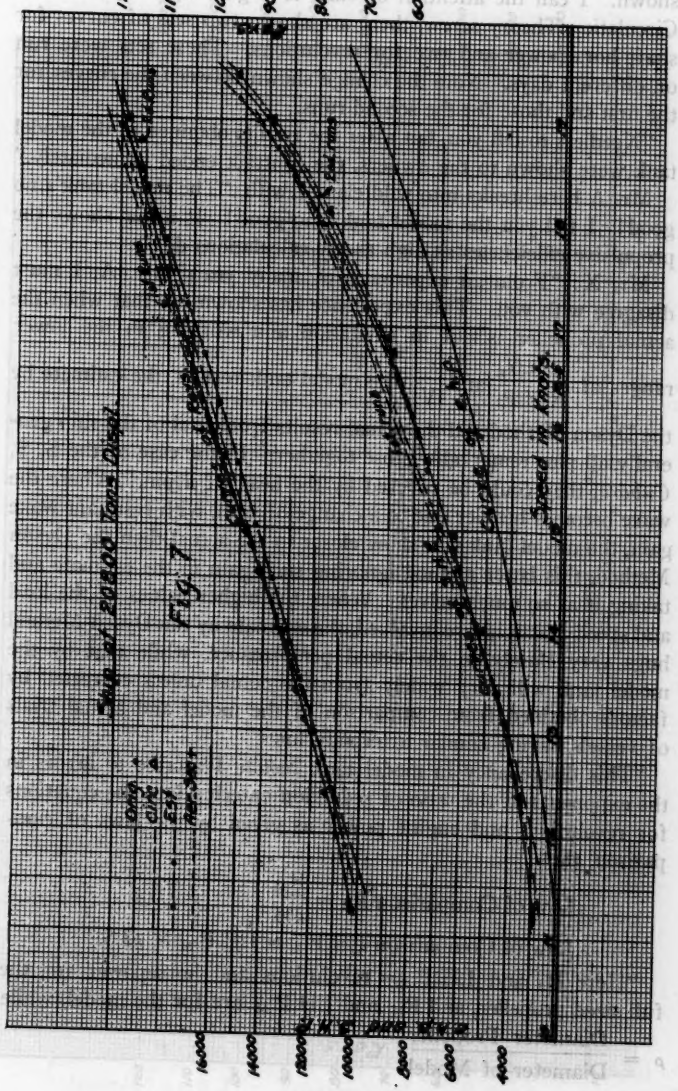

120
110
100
90
80
70
60
Speed in Knots
16000
14000
12000
10000
8000
6000
4000

FIG. 7

shown. I call the attention of such to Figure 7, where, for the Circulation Theory screw, there are shown two sets of curves for shaft horsepower and two for revolutions. These sets were run on different days. Note the change shown between the curves for the first and those for the second runs.

"Needless to tell you, but after the results obtained in the model tank were shown to the owners, the contract ended immediately."

Mr. A here breaks into wild clamor with, "The model tank's no good! I tell you the model tank's no good! It never shows the propulsive efficiencies as high as my calculations do!"

Mr. X: "From an experience of many years, Mr. A, I must disagree with you. This experience has shown me that when the actual propeller is not in cavitation and working in the lower ranges of $\frac{\text{e.h.p}_s}{\text{E.H.P.}}$ and $\frac{v}{V}$, the model tank results, the estimate by the Dyson Method and the actual results from trial of the ship generally all plot remarkably close together. In the case of the S. S. *Cubore,* they could be covered by the head of a pin. Where the wake becomes heavier due to a change of trim, with resultant wake gain, the model tank will show closer to the actual than the Dyson Method, this latter method ignoring such gain in the estimate and taking it as so much velvet. Where the higher values of the load and speed fractions are involved, the estimate by the Dyson Method hugs very closely to the actual performance, while that by the model tank will vary widely from the actual unless corrected by factors obtained from comparison of the model and actual trials of vessels having similar afterbody lines.

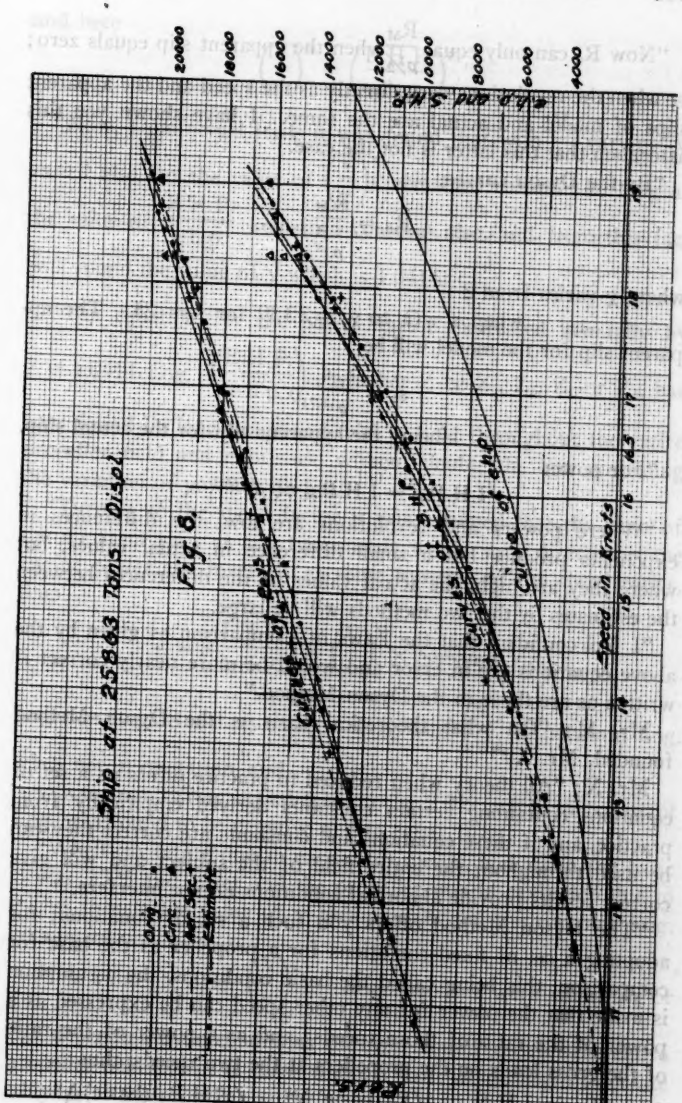
"This discrepancy in model tank results I ascribe to errors in the exponents of the ratio of reduction entering into the equations for revolutions and engine power as given by the laws of comparison, thus

$$\text{Law of Comparison } R_d = \frac{R_M}{\rho^{1/2}}$$

$$\text{Law of Comparison I.H.P.}_d = \text{I.H.P.}_M \times \rho 3.5.$$

"Here R_d and I.H.P.$_d$ = revolutions and horsepower for the full sized propeller, and R_M and I.H.P.$_M$ are for the model, while

$$\rho = \frac{\text{Diameter Propeller}}{\text{Diameter of Model}}.$$


Ship at 25863 Tons Displ.
Fig 6.
Curves of Slip
Curves of I.H.P.
Curve of Speed
Speed in Knots
I.H.P. and S.H.P.
Revs.
Actual
Ac. per Sect.
Estimate

"Now R_d can only equal $\frac{R_M}{\rho^{\frac{1}{2}}}$ when the apparent slip equals zero; or when the revolutions vary directly as the speed and the apparent slips of model and actual are the same. I have shown you this afternoon that this latter is not the case.

"By the Dyson curves,

$$R_d = \frac{R_M}{\rho^{\frac{1}{x}}}$$

where χ varies from 2

when $s = 0$, to $\chi = 2.139$ for $s = .25$. The apparent slip for the model will be

$$s_1 = \frac{s}{\rho^{.1271}},$$

s being the apparent slip for the actual ship.

"For power

$$\text{I.H.P.}_d = \text{I.H.P.}_M \times \rho^{3.5621}.$$

"Where powers and revolutions are low, the differences in exponents produce but a small difference in actual values, but where they are high, the actual values of the differences between the estimates by the two methods will be large.

"I feel confident that the Laws of Comparison as given by the above equations are in error and would be more nearly correct if written as are those in the Dyson Method."

Mr. M: "On what theoretical basis is the Dyson Method founded, Mr. X?"

Mr. X: "All theory when reduced to practice offers us a set of equations containing certain constants derived empirically from practice, and if these equations and constants are correct they can be used throughout the entire field of the subject and will give correct results provide abnormal conditions do not interfere.

"The Dyson Method offers you such a set of equations; for adopting a set of fixed conditions for a propeller as the base for comparison, this being called the basic conditions, the assumption is made that the power for any other speed can be expressed as a power of the ratio of those speeds; and as a power of the ratio of the gross loads on the propellers at the two speeds, thus:

$$\frac{\text{Basic I.H.P.}}{\text{Actual I.H.P.}_d} = \left(\frac{V \text{ (basic)}}{v \text{ (actual)}}\right)^w = \left(\frac{\text{E.H.P. (basic)}}{\text{e.h.p. (actual)}}\right)^y,$$

and here

$$\left(\frac{V}{v}\right)^{w} = \left(\frac{E.H.P.}{e.h.p.}\right)^{y}."$$

"Now, from actual trials it was found that with $\frac{V}{v}$ varying, w varied also but the value of $\left(\frac{V}{v}\right)^{w}$ remained constant so long as the value of $\frac{E.H.P.}{e.h.p.}$ remained constant also, and from this fact was found the value of y which is 1.0414.

"The equation for the power factor expressed generally by $Z = 1.0414 \text{ Log. } \left(\frac{E.H.P.}{e.h.p.}\right)$, whether Z be Z_p, Z_g, Z_1 or Z_c, has been found by long usage and trial to hold throughout the entire propeller field and expresses, without doubt, the law governing the variation of power with load.

"Turning to the speed factor it was assumed that the ratio of the apparent slips at any two speeds would vary directly as the engine power and inversely as some function of the speed ratios, thus

$$s = S \frac{I.H.P._d}{I.H.P.} \times \left(\frac{V}{v}\right)^{\chi} \times \text{constant, or}$$

$$\text{Log. } s = \text{Log. } S - \text{Log. } \frac{I.H.P.}{I.H.P._d} + \chi \text{ Log. } \left(\frac{V}{v}\right) + \text{constant Log.}$$

when not in cavitation, and

χ was found from actual trials to be 2.869, while the constant Log. was found to be negative and to equal .0788.

"When cavitating due to excess tip speeds, and when the basic S was .1325 and lower and the propeller was in cavitation due to excess thrusts the value of χ became 3.09 and the constant Log. became zero.

"The values χ Log. $\left(\frac{V}{v}\right)$ + Log. constant were called Z_s or Z_{sc}, depending upon conditions and were found to cover the entire propeller field except in cases where a strut would begin to create eddying at the propeller as the squat of the vessel increased with

the speed and as the amount of eddying and reduction of pressure at the propeller became increased with the increasing squat.

"The equations for Z_s express the law by which speed affects apparent slip and the equation for the apparent slip undoubtedly expresses the law governing the combined effects of engine power and ship's speed upon that quantity.

"If these assumptions, together with those I have already given concerning the causes of cavitation, form a theory, well and good. If they do not, then let the theorist back track until he unearths the underlying theory. It is as easy to travel from Washington to New York as it is to travel in the opposite direction.

"Now, gentlemen, I wish to say in

CONCLUSION

that the kernel of the propeller problem nut is the afterbody of the ship's hull with its effect upon wake and power loss. Knowledge of this can only be gained by experience, more experience and then still more experience. Yet even with this qualification failure may be encountered, particularly with what may be termed border line vessels.

"The true line to follow in the attempt to increase propulsive efficiencies is not that of attempting to develop new theories for the calculation of unimportant details, but is one in which the Naval architect is particularly interested. It is the improvement of afterbody lines of hulls. In his search for this he must realize that those lines which tow the easier in the model tank are not necessarily the easier to drive with a propeller. What is gained in ease of towing may be more than lost by extravagant increases in the power loss factor.

"No man can mount himself upon his pet theoretical hobby, armed only with the lance of self-assurance and backed up by his henchman, inexperience and expect any better success in his attacks on the propeller problems, than did Don Quixote have in his encounter with the windmill."

The meeting now breaks up, some of the participants as they departed stating, like the Chinaman, that what was good enough in the way of design methods for their fathers and grandfathers was good enough for them, while one of the Circulation Theory advocates was heard to exclaim, "Thank God for the Circulation Theory for it is a blessing to mankind."

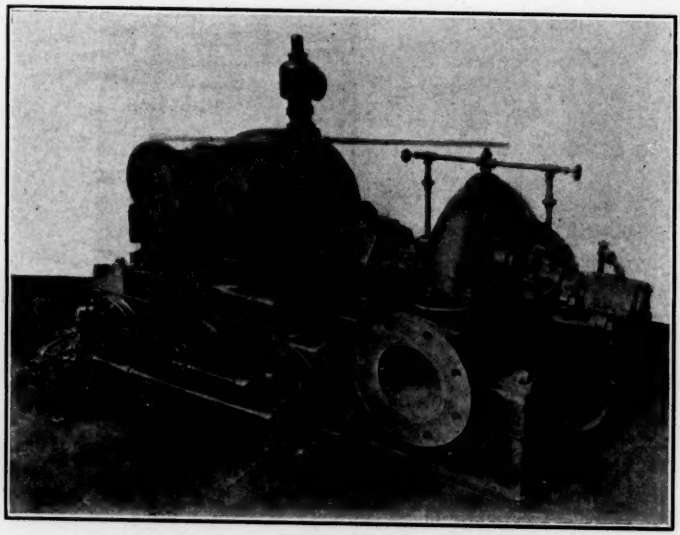


FIGURE 1.

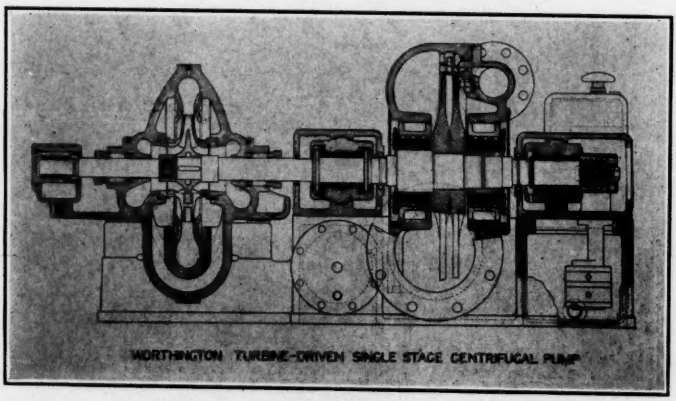


FIGURE 2.

WORTHINGTON SINGLE STAGE CENTRIFUGAL PUMP.

By J. B. Lincoln, Mechanical Engineer, U. S. Naval Experiment Station.

The increasing use of higher steam pressures has resulted in the introduction of high pressure centrifugal pumps for boiler feeding purposes. In a majority of makes the manufacturers have added more stages to their line of pumps or have installed two or more pumps in series to meet the service required. At least one manufacturer has developed a centrifugal pump with a single impeller to meet the demand for moderately high pressure boiler feed service.

The following article describes and gives results of tests of a Worthington turbine driven single stage double suction centrifugal pump. This pump was designed for boiler feeding at a pressure of 600 pounds per square inch with a rated capacity of 650 gallons per minute. The normal full load speed is 6000 R.P.M.

The unit, comprising pump and turbine mounted on its base is shown in Figure 1. The weight of the unit as shown is approximately 3600 pounds. The approximate overall dimensions are: length 5 feet, width 2⅔ feet and height 2⅓ feet. The pump suction and discharge connections are for 6-inch and 4-inch pipe, respectively. The steam connection to the turbine is for 3-inch pipe while the exhaust opening is for 8-inch pipe. The pump impeller and turbine discs are mounted on the same shaft, which is mounted in three bearings, one at each end and one between the pump and the turbine. The sectional assembly of the unit is shown in Figure 2.

The pump impeller, which is made of bronze, is 10½ inches in diameter and is held on the shaft by means of a key and a threaded collar. The water discharged from the impeller passes through a specially designed diffusion ring directly into the casing. The

casing, which is made of cast bronze, is split in a horizontal plane through the center line of the shaft. The lower half of the casing which carries the suction and discharge connections is mounted on one end of the base.

The double suction comprises two converging passages leading from either side of the impeller to the single suction connection, according to conventional practice. No sleeves are provided on the shaft where it passes through the stuffing boxes of the suction passage. The outboard bearing on the pump serves to support the extension of the pump shaft.

The sealing rings which separate the discharge chamber from the suction chamber and surround the impeller hubs are of special design to give adjustable and labyrinth construction in a simple manner. The combination of a radial labyrinth with a horizontal sleeve type construction provides effective means of preventing leakage. The design permits of ready adjustment. Leakage is prevented along the shaft by the use of soft packing separated by water sealed lantern ring and brass follower ring adjusted by two nuts on each ring.

The turbine casing is also divided on a horizontal plane through the shaft center line. The steel steam chest section is split vertically and is bolted to the upper half of the casing. The lower half of the casing which carries the exhaust connection is mounted so that both halves of the casing are free to expand laterally away from the shaft.

The two velocity stages of the turbine incorporate two separate forged steel wheels shrunk onto the shaft against a shoulder and further held by a threaded collar. The steam buckets are of stainless steel and are double riveted for maximum strength. The reversing buckets are mounted on a segmental ring held in the proper position in the casing with respect to the wheels. The steam nozzles are machined from a steel forging and are bolted to the steam chest. Eleven of the thirteen nozzles open directly from the steam chest. The other two nozzles are controlled by a single valve for use at the higher pump loads or with reduced steam pressure.

The shaft glands are of the three ring carbon type with garter springs. Each ring is mounted in one of three grooves machined

in cages mounted concentrically with the shaft between the two halves of the turbine casing. The middle groove in each case is vented with suitable pipe openings. Guard rings with radial webs are fixed to the shaft close to the shaft glands to prevent the possible entrance of foreign material.

The three bearings which carry the turbine and pump shaft are of the forced feed lubricated type. The oil is furnished by a pump mounted in the base and which is driven through a worm and gear from the main shaft. The oil used for lubrication is held in the base, having a capacity of approximately 16 gallons. There is a pipe coil within the base to permit of the circulation of water for cooling the oil. Oil from the bearings is returned through suitable pipes to the reservoir in the base.

Oil is supplied to the bearings at a normal pressure of about 5 pounds per square inch. Should this oil pressure fail for any reason an automatic device provides for stopping the turbine by tripping the main steam valve. Each of the bearings is separated from the pump and turbine casings and are bolted to the base independently. The presence of the quantity of oil in the base provides for a uniform temperature for the pump and turbine supports. This together with the provision for free expansion of the turbine and pump casings reduce the possibility of misalignment of the bearings to a minimum.

The turbine is provided with a constant pressure governor and an emergency overspeed trip. The constant pressure governor is designed to maintain a fixed pressure at the pump discharge. This governor is operated by oil under pressure furnished by a rotary pump driven from the same vertical shaft that drives the pump for lubricating the bearings. The oil from this pump operating through a pilot valve operated piston controls the main steam valve. Adjustment of the oil pressure by means of a suitable by-pass valve in the oil line provides for operation over a range of pump discharge pressures. The oil pump is immersed in and takes its suction from the oil reservoir in the base of the unit. After passing through the regulating device the oil is returned to the reservoir.

The emergency trip is of the centrifugally operated type. As the speed reaches a set maximum a plunger is thrown out, opening a

small valve which in turn relieves steam pressure from beneath a piston. This piston then moves to shut off the flow of steam to the turbine.

The pump unit was installed for test at the Engineering Experiment Station, Annapolis, Md. Data and information were obtained to determine the suitability of the unit for boiler feed purposes in the Naval Service.

In the conduct of the test one of two 5500-gallon calibrated tanks were used as a source of supply from which the pump took its suction. The pump delivered the water through a Venturi meter, thence through heaters into a second 5500-gallon tank. Valves were located in the suction and discharge lines and were used during the tests to vary conditions as desired. The two tanks were interconnected through suitable valves and fittings so as to reverse the flow when one of the tanks was emptied. The Venturi meter used to measure the quantity of the water delivered by the pump was located so that there was at all times a positive head on it. Special precautions were taken to see that all air was removed from the piping leading from the Venturi entrance and its throat to the mercury manometer used to obtain the pressure drop through the meter. The Venturi meter was checked before the tests against the calibrated 5500-gallon tanks. The pump discharge pressure was obtained from a pressure gauge located in a fitting placed between the pump and the discharge valve. A mercury manometer was attached to the suction pipe four feet from the pump. With a full tank of water a static head of approximately fifteen feet of water could be had at the suction to the pump. With the use of the valve in the suction line it was possible to vary this head within certain limits as desired. The temperature of the water was obtained from a thermometer located in the suction pipe.

Steam to the turbine was supplied from the main steam line through a 3-inch pipe. Between the main line and the turbine control valve were located a hand operated regulating valve, a throttling calorimeter, a pressure gauge and a thermometer. The steam pressure in the nozzle box was obtained from a gauge located at that point. The exhaust from the turbine was conducted to a surface condenser from which the condensate was

pumped to weighing tanks mounted on calibrated platform scales. The exhaust pressure was regulated by a valve located in the pipe leading from the turbine to the condenser.

The speed of the unit was obtained from readings taken on a positive counter geared through a suitable reduction from the main shaft. Instantaneous readings of the speed could also be obtained from a tachometer inserted into one end of the counter shaft.

The principal data obtained during the tests comprised the determination of the pressure-capacity characteristics at various speeds, the steam consumption of the unit, the operating steam pressures and the performance of the pump at various water temperatures and with various positive and negative heads at the pump suction. Economy characteristics of the unit were deduced from the above data. Operation of the unit under control of its governor was also undertaken to determine its effectiveness in maintaining constant discharge pressures.

The curves on Figure 3 show the head-capacity characteristics of the pump at speeds of 3000, 4000, 5000 and 6000 R.P.M. During these runs the speed of the unit was maintained constant while the valve in the discharge line was adjusted to vary the capacity and pressure. The runs varied in duration from 2 to 20 minutes. Readings were taken at intervals of two minutes. The curves were plotted from average data with proper corrections. The shut off points on each of the constant speed curves were obtained from actual data with the discharge valve completely closed. All of the curves have a reasonably flat pressure characteristic from no delivery up to 750 or 800 G.P.M. The maximum pressure reached at 6000 R.P.M. was 615 pounds per square inch. At this pressure and speed the delivery was 615 G.P.M. It will also be noted that the pump conservatively meets its rated capacity of 650 G.P.M. at 600 pounds per square inch pressure. It would probably be satisfactory to operate the pump up to 750 G.P.M. with cold water. The variation in pressure at constant speed is not greater than approximately 10 per cent from shut off up to 750 G.P.M. During the runs made to obtain the data for these curves the valve in the suction line to the pump was adjusted to maintain a zero head. The water was not heated during these runs except for the heat added by the pump. In most cases the temperature of the water did not exceed 100 degrees F.

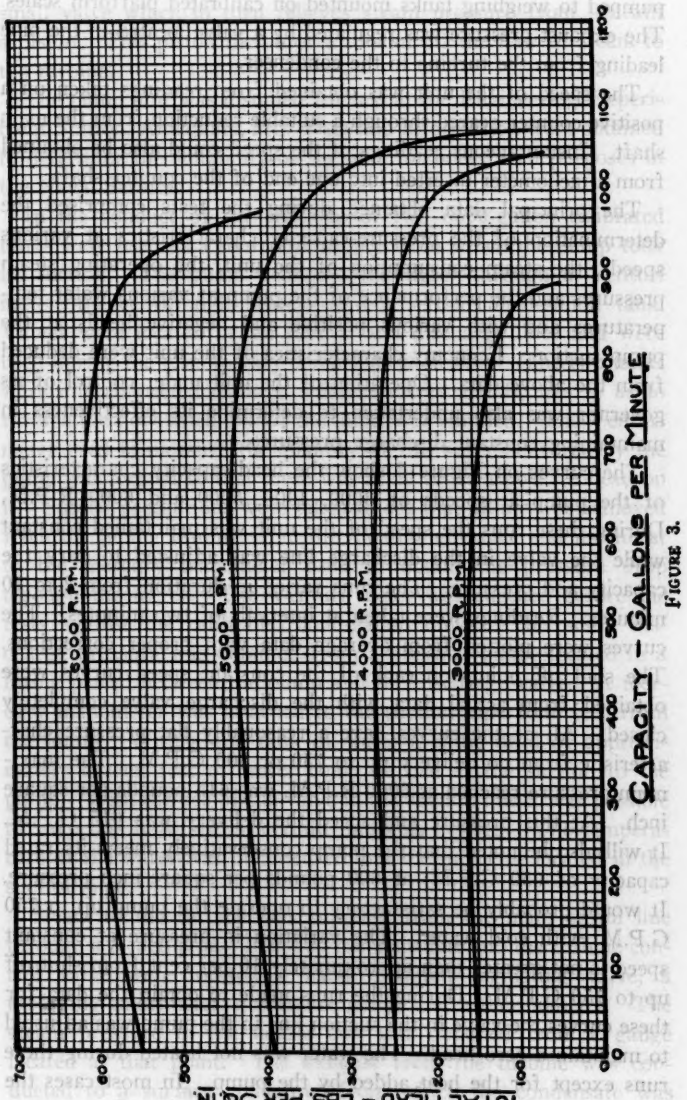

TOTAL HEAD - LBS. PER SQ. IN.
CAPACITY - GALLONS PER MINUTE
5000 R.P.M.
5000 R.P.M.
4000 R.P.M.
3000 R.P.M.
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
100
200
300
400
500
600
700

FIGURE 3.

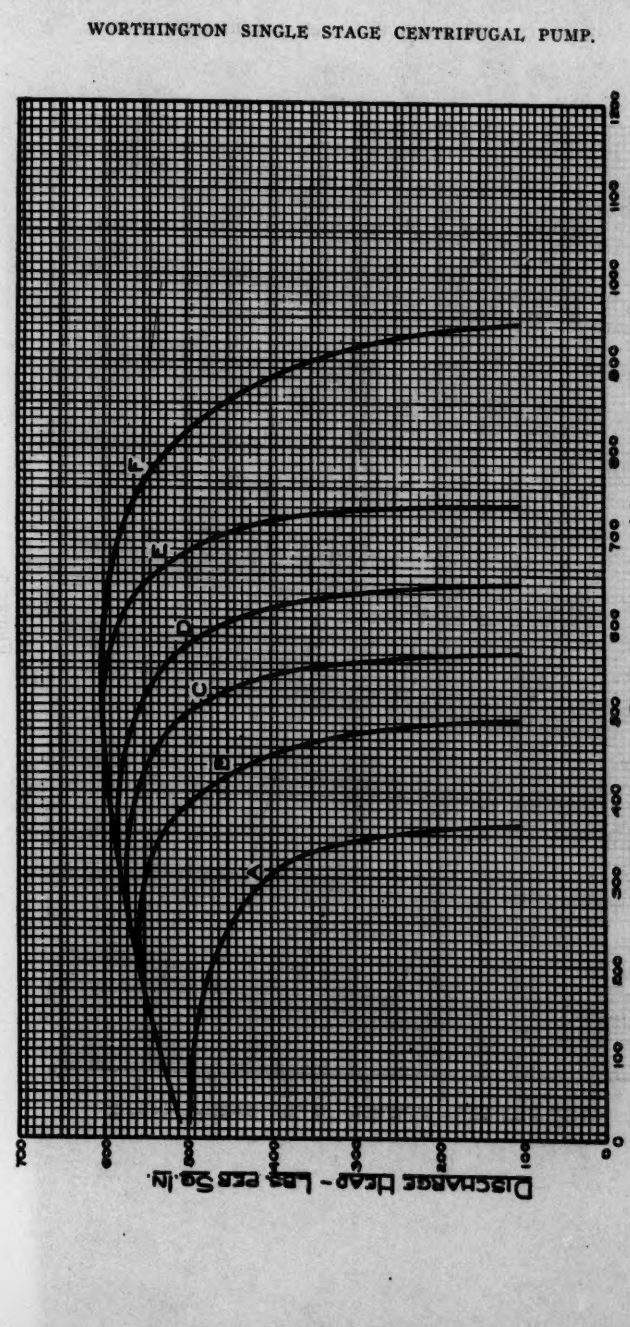

Discharge Head - Lbs. per Sq. In.
Capacity - Gallons per Minute
A
B
C
D
E
F

FIGURE 4.

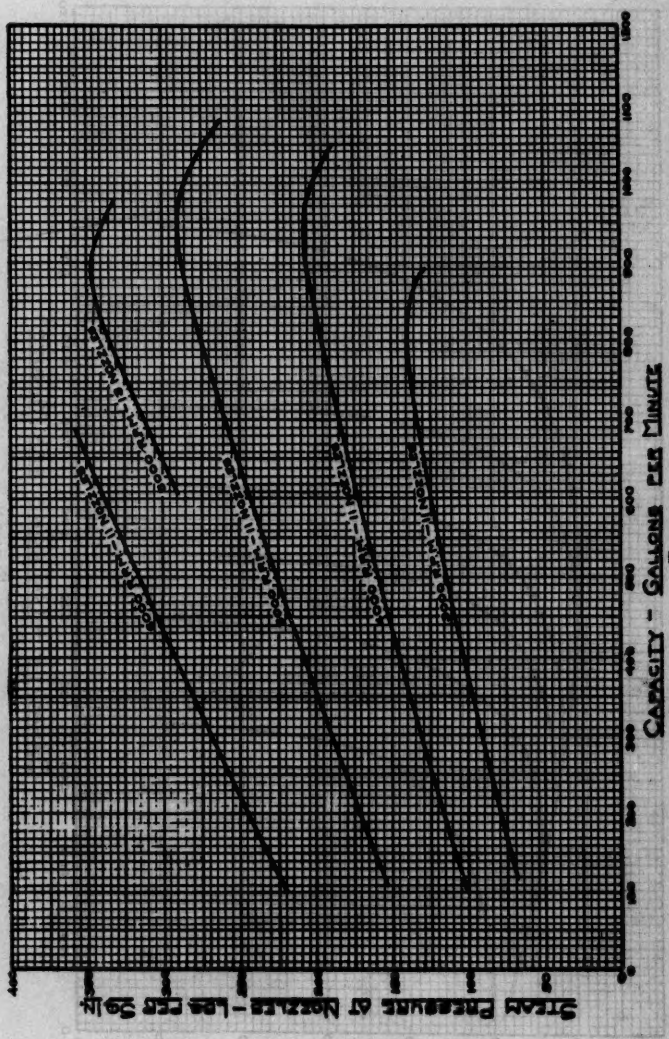
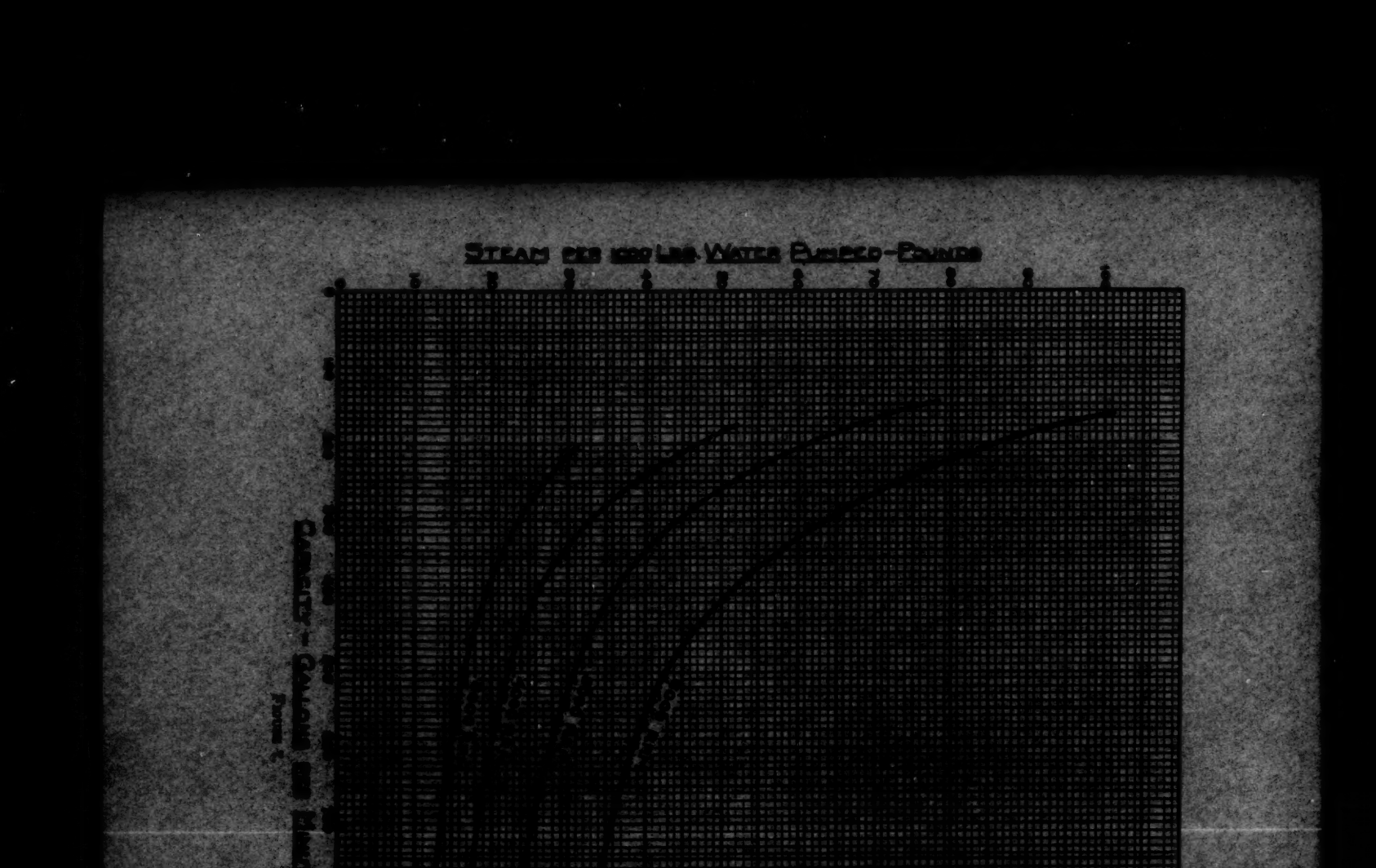

Capacity - Gallons per Minute
Steam Pressure at Nozzles - lbs per Sq. In.

FIGURE 5.

Steam per Water Horse Power – Lbs. per hour
Capacity – Gallons per

Figure 4.


Steam per 100 Lbs. Water Burned-Pounds

The curves on Figure 4 show the effect of the pressure-capacity characteristics of varying the pump suction. The data for these curves was obtained with the water heated to a temperature of 160 degrees F. and with the pump operating at a constant speed of 6000 R.P.M. The various suction conditions were as follows:

Curve A—12½-foot lift	Curve D—4½-foot lift
Curve B—11¼-foot lift	Curve E—0 head
Curve C—8-foot lift	Curve F—7¼-foot head

Data relative to the steam pressures required to operate the unit are given in the form of curves on Figure 5. These curves were plotted from steam data taken during the runs made to determine the characteristics shown in Figure 3. The turbine exhaust was maintained at atmospheric pressure during these runs. It will be noted that a minimum nozzle pressure of about 325 pounds per square inch is required to operate the unit with a capacity of 750 G.P.M. at a discharge pressure of about 600 pounds per square inch. Operation under these conditions requires the use of all of the thirteen turbine nozzles. The steam pressure drop through the governor valve varied from 6 to 20 pounds per square inch, depending on the load. Under full load conditions on the pump this would require a minimum pressure of about 350 pounds per square inch to operate the unit.

The steam consumption of the unit based on delivered water horsepower is shown graphically on Figure 6. The curves on this figure show a remarkably low steam consumption of approximately 48 pounds of steam per water horsepower under full load conditions. The figures for the 5000 and 6000 R.P.M. runs on this inclosure coincided so closely that a single curve has been drawn.

The steam economy based on weight of water delivered is shown by the curves on Figure 7. The data for these curves as is the case with Figures 5 and 6 must be considered in conjunction with the characteristic curves on Figure 3, that is, as the capacity is changed at constant speed the pressure changes follow the characteristic curves.

The tests made to determine the effectiveness of the governor to maintain a constant discharge pressure on the pump show that the governor was capable of holding the pressure within a variation of 10 per cent from no delivery to a maximum of 650 G.P.M. Figure 8 shows a characteristic curve of the regulation obtained together with the variation in the nozzle steam pressure. The governor was set for a delivery pressure of 500 pounds per square inch at a capacity approximating 300 G.P.M. From this point the capacity was increased to 125 per cent of full capacity and then decreased to no delivery and subsequently increased to the initial 300 G.P.M.

The operation of the unit was particularly free from vibration at all speeds up to 6800 R.P.M., which was the maximum noted during the tests.

XC

PRESSURE — LBS. PER SQ IN.
700
600
500
400
300
200
100
0
0
100
200
300
400
500
600
START
PUMP DISCHARGE PRESSURE
PRESSURE BEFORE GOVERNOR
NOZZLE PRESSURE
CAPACITY — GALLONS PER

Figure 9

A – Increasing Capacity
B – Decreasing Capacity
600
700
800
900
1000
1100
1200
Gallons per Minute

A REVIEW OF "PRACTICAL APPLICATION OF MODERN HYDRODYNAMICS TO MARINE PROPULSION."

(FEBRUARY, 1927.)

BY COMMANDER J. M. IRISH, U. S. NAVY, MEMBER.

As long as propellers maintain their present dominant position in marine work there will undoubtedly continue to be developed new theories of propeller action. The form of these methods will vary with the experience and the physiological reaction of the investigation to the data available.

All theories of propeller design may be classed as follows:

(1) those based on pure theory.
(2) those developed from tests of model screws.
(3) those developed from results of actual trials of ships.

Irrespective of the basis of the various theories, they all are alike in one respect when used for actual design methods: viz., that in applying the theoretical conclusions to actual conditions certain arbitrary or empirical constants must be used to bring these results in line with actual test reports. When any variations from standard hull conditions exist the degree of success of any system of propeller design depends on the accuracy of these conversion factors.

Prof. Slocum's article comes under group one. From purely theoretical assumptions and by various mathematical steps he finally arrives at two expressions, one for torque and one for thrust. The author's original work consists of evaluating the constants and developing means for applying them to various blade forms. The circulation theory has received more attention in Europe than in America, and Prof. Slocum is to be congratulated on bringing these theories to a more practical and useful basis.

It is to be regretted that so much data has been omitted that would throw more light on the method of applying these formulas, and give more confidence in their application. Thus the reader is left with no idea of the values of the constants K_1 and K_2 which the author states are "empirical factors which must be determined experimentally." Another incomplete but interesting subject is the determination of "wake." As the speed used in the equations is speed corrected for wake, this wake correction is essential for accuracy, and the author states that the wake as determined by his method "is far more accurate than the value of shaft horsepower specified." But no hint is given of the method to be used other than "it is sufficient to establish a relation between effective wake and relative dimensions of the propeller and afterbody of the vessel."

Another variable which is left indeterminate is the angle of incidence "B." This may be varied between 2 to 8 degrees and according to the author "must be of a certain definite amount depending on the load of the propeller." As the revolutions and slip are dependent on the assumed value of "B," more accurate knowledge of how to fix it is essential.

It is evident therefore that the essential conversion factors have been omitted from this article, and the practical value of the method cannot be estimated until they are further elucidated.

Designers of propellers are faced with various problems which can be classed as follows:

(1) With given hull lines, e.h.p. curve, desired speed, and limit to diameter, find best type of wheel, area, diameter, pitch, R.P.M., and S.H.P.

(2) With given hull lines, e.h.p. curve, desired speed, limit to diameter and limit to R.P.M., find best diameter, area, pitch and S.H.P.

(3) With given hull lines, engine H.P. and R.P.M., e.h.p. curve, limit to maximum diameter, find best area, pitch and best speed which may be obtained.

(4) Same as (3) but desired speed is designated.

In paragraph (20) the author has outlined his method of design assuming conditions as in case (4) above, that is; he has the engine power and R.P.M., the lines of the vessel which limit the propeller

diameter, the designated speed, and the e.h.p. curve. He then states "The propeller must be so designed, therefore, as to develop the specified torque, M, at the specified speed of revolution, N."

If the problem were only that, propeller design would be comparatively simple. The real problem is to design the propeller so that when operating behind a given hull it will absorb the specified torque at the specified speed of revolution *and will deliver the required thrust at those revolutions.* This delivered thrust ensures the speed V through the water.

Many a propeller absorbs the torque at the given R.P.M., but fails to deliver the thrust needed, as designers well know to their cost and chagrin. Furthermore, an able designer may be able to obtain, under given conditions, an even more efficient wheel and thereby increase the maximum speed, a "consummation devoutly to be wished."

It does not appear that the two equations for torque and thrust lend themselves readily to the solution of these problems, nor to the *best* solution without considerable cut and try. This is further complicated by having six types of propellers from which to choose, more than one of which should at least be tried to insure that the best wheel is selected.

It is interesting to note that no reference is made to determining the pitch or slip. This relationship is apparently involved in the various empirical factors mentioned above and by means of the equations the values of torque, thrust, diameter, revolutions and speed are so related that pitch and slip do not enter. If this is so, it makes a most interesting difference between this theory and any other which has yet been developed.

As the author apparently intends further to develop the practical applications of his theory many of these defects may be eliminated, but as it stands it is of no practical value to the designer who is seeking after means of increasing the propulsive efficiency of the screw propeller.

As this article is devoted entirely to theory, it is interesting to compare it with an article by Prof. Slocum in the February, 1926, issue of "Marine Engineering and Shipping Age." The author there describes the installation of a propeller designed by his method on the S. S. *Challenger,* whereby with the same horse-

power the sea speed was increased from 10.4 knots to 10.83. His propeller had a variable pitch and larger projected area than the original.

In the April issue of the same magazine, Admiral Dyson in criticizing this article, showed how by use of his method it was apparent that the original wheel was in cavitation due to excess thrust, and, by using the same data that was available to Prof. Slocum, that it was perfectly possible to design a propeller of standard blade profile with constant pitch which would give equally good results as those obtained by the propeller designed by the circulation theory. He further showed that the gain in efficiency was not due to anything inherent in a blade designed by the circulation method, but was simply due to increasing the area and decreasing the mean pitch which lifted the propeller out of cavitation.

Until those interested in the circulation method of propeller design can show some greater improvements than those which can be obtained by the present standard methods, it is difficult to see any reason for adopting that system, especially as the cost and difficulty of manufacture are in favor of the present standard designs.

Note: In connection with this discussion it may be of interest to read a discussion by Dr. Bauer of a similar subject reprinted under the Notes entitled—"The Application of Circulation Theory to a Hydraulic Propeller by Dr. H. Boerck."

FLYING AND THE NAVAL ENGINEER.

By Assistant Secretary of the Navy for Aeronautics, Edward P. Warner, Member.

Mr. Toastmaster, Members and Guests of the American Society of Naval Engineers:

It is, it seems to me, not inappropriate that the Society of Naval Engineers should invite a representative of aeronautical engineering, and one connected with the Navy's work in the aeronautical field, to talk on a subject at least partially aeronautical, for the kinship between the problems of the aeronautical scientist and of the designer of aircraft and those of the naval constructor and the naval engineer has been very close; and the similarity of their problems, and the manner in which the work done by one group may prove of benefit to the other becomes more and more apparent as time passes, and as aeronautical engineering becomes progressively more clearly and more exactly one of the engineering sciences, with an increasing mass of data based on research and experience upon which the practitioner of that particular branch of the engineering art may draw. There is obviously much of parallelism between the craft that operates in the air and that which operates on the surface of the sea. The naval constructor and aeronautical engineer alike are concerned with vehicles, operating in and through fluids, and must base their designs, so far as the exterior form of the craft is concerned, upon a knowledge as exact as can be obtained of the laws that govern the relation between the motion of fluids and their resistance to the motion of bodies passing through them. Both use the same method of propulsion, the screw propeller, devised for surface craft long before there was any experience with airplanes and airships, and which now takes its place in aircraft as well, and the experience of the marine engineer has been called upon in making the application. The aeronautical engineer and the naval constructor meet with a kin-

dred problem, too, in that the complexity of the relationship between the structures which they design and the fluids through which these structures move is such that the forces acting, in their distribution and in their magnitude, are complex in the extreme, and the calculation of the parts of the structure to sustain the loads so imposed in virtue of the relative motion through the fluid is correspondingly complex. But finally, and this is most important, the naval engineer or the naval constructor, the engineer whether he works with the hull or with the machinery, and the aeronautical engineer whether he is concerned with the structure or with the power plant or with any particular part of the craft that goes into the air, are alike in that they are continuously and intimately concerned with the economy of space and economy of weight; the economy of space most significant in the one instance, economy of weight in the other, but both of real importance both on the sea and in the air, standing apart from some other branches of the engineering profession where space is of very secondary importance, and weight of no importance at all, except as it involves the use of an added amount of material, and consequently added expense in getting the material. This relationship between the engineering of ships and the engineering of aircraft has been one of mutual benefit since the days when flying first began. The advantage that aeronautical engineering has derived from the experience of the naval engineer and of the naval architect is too well known to all naval officers here for me to dwell upon it. It is too well known to require emphasis here, how large a share of the design of the modern airplane and how many of the features that enter into every airplane are based upon the work of the naval architects, and specifically of naval officers, engaged in engineering and in construction.

There are some of those here tonight who have had a large share in that application, and whose own researches have been largely influential in determining the form that the seaplane and the flying boat take today. But I say that the relationship has been one of mutual benefit, and that while the design of the airplane has gained in great measure from marine experience and has been based in considerable part upon that experience, so too the naval engineer, and indeed all marine engineers may have something to gain from

keeping watch on the further development of aeronautical engineering. There is something for you to gain from aeronautical engineering for several reasons, and first because it is often convenient in performing our experiments on full scale models, in making our service trials, to be able to go at once to an extreme beyond anything which we are likely to desire to put to practical use. If we want to know in aeronautical engineering what can be done with an airplane of moderate size and of moderate power, it is often expedient and useful to start out by seeing how small a machine can be built and what is the lowest power that makes it possible to get into the air at all and it is useful for the marine engineer to keep watch over what can be done in aviation, where the limit of weight economy is necessary and where studies on elimination of ounces must be carried to an extreme. Furthermore, there are practices and methods which are of present or potential use in marine, and more especially in naval engineering which had earlier found a place in aeronautical engineering by virtue of the spur of necessity, and the spur of necessity has required too the development of theories for applications to airplanes which subsequently have found applications to the propulsion of surface craft. The airplane serves then as a useful vehicle of experiment, not only for the design and building of new airplanes, but to furnish data for other branches of engineering as well, and specifically for this one with which you are concerned.

Although the screw propeller had been used for the propulsion of ships for many years before an airplane was built, and although propellers had been used in some form more or less crude for nearly, if not quite, a century before the date of the first flight, there was, I think it is quite safe to say, a more extended development of theory of propeller design, theory on which propeller design could be properly and directly based, in application to the air propeller during the first 15 or 20 years of flight than in application to the marine propeller during all that century, for the very obvious reason, among others, that whereas almost any propeller suffices to drive a ship after a fashion, one having a rather high efficiency had to be made before the airplane could get into the air at all. The line between success and failure was very sharply defined, and correspondingly intensive study had to be

given to the basic principles of propeller operation, a study which has I think made substantial contributions to the design of the marine propeller in the last decade. A hint of its importance is found in your award of a gold medal to Dr. Slocum tonight for a paper on marine propulsion, a paper based very largely on aerodynamic theories originally developed with sole reference to aircraft. Another example of that same spur of necessity is in the superlative importance of a sufficient degree of strength in the aircraft to insure it against any failure, a degree of importance no greater to be sure than that in the hull of a ship, but subject always to limitations of weight and of bulk far more severe than those which fall upon the designer and builder of a vessel operating on the surface of the sea. And in the light of that necessity, there has been again development of methods of research and experiment, and of measuring instruments, first produced with no other thought beyond application to aircraft, but subsequently extended to other fields as well, and specifically to yours.

If we consider airplanes, ships, or locomotives, and motor trucks, and determine their cost of operation, it is easy to calculate upon a sufficient number of assumptions a figure which corresponds to the cash value of weight saving in the structure, that is, a figure which represents the additional amount of money that it is profitable to spend in order to reduce the weight of the structure by one pound or any other amount that you like; and if we do that for a merchant ship we find that the cash value of saving a pound of weight comes out at a very few cents indeed, whereas in the airplane it runs up to something on the order of twenty-five or thirty dollars per pound saved. That is, the cost of the material is of no importance. Even on economy grounds alone, it is worth while using material that costs anything within reason, five, ten, or fifteen dollars a pound in order to save a minute percentage in weight. So again the airplane serves as a laboratory, and furnishes an incentive for the development both of new methods of construction with primary reference to weight economy, and a spur to the further development of new materials and especially in recent years, to the production of the lighter metals and alloys. A great British metallurgist remarked not very long ago that we had lived through the ages of stone and of bronze and of iron, that we

were now in the age of steel, and that the age of aluminum was just around the corner ahead. If that be true, if we are to pass into an era where the use of lighter alloys will become more and more important in all branches of engineering construction, the airplane industry and the research laboratories, which have applied their efforts with specific reference to aeronautics can claim a very large part of the credit. In building airplanes we do not have to worry about costs of material very much, and in building naval craft there is at least a much greater degree of freedom from that particular type of economy pressure, a greater degree of freedom from worry about unit cost of material, or of refinements in construction method which will bring a real increase in efficiency in the work for which naval craft are designed than can exist in the design and construction of merchant vessels.

So it is doubly fortunate that aeronautics has taken an important place in the naval service. It is fortunate for the development of the airplane, because it has concentrated upon that development the attention of naval officers and of engineers outside the service concerned with the construction of naval material, and the airplane has profited greatly thereby. It is fortunate too for the Navy, not only for the benefit that the service can derive from the use of the airplane with the fleet at sea, but also for the lessons that the designer of every kind of naval material may be able to draw from aeronautical experience within the Navy and without. You here are better qualified than I can be to say to what extent the benefit has as yet been felt in the design of engineering structures of ships. I know that it has been of great benefit to aeronautics. And I am confident that a continuance of that development in the same way and through the same channels that have attended upon it during the last 10 or 15 years will bring continued benefit to all parts of the service, both in the operation and in the design and production of material.

THE "SPANISH PRISONER" PROBLEM.

By Lieutenant (J. G.) Leonard Kaplan (C. C.), U. S. N., Member.

The "Spanish Prisoner" or $x + y$ problem is the problem

$$x^2 + y = 7$$
$$x + y^2 = 11$$

It has long been known as a trick problem, receiving considerable attention in the last decade of the past century. However, someone frequently brings it around now for solution, so that it is well to know the problem and appreciate its difficulty before attempting to solve it.

The solution which I have prepared below does not concern itself with the geometric properties of conic sections, but merely presents a straightforward algebraic solution (with a single digression to trigonometry for the solution of a cubic) of what at first glance, appears to be nothing more than a simple problem in algebra.

No use has been made of factoring for the obvious roots, $x = 2$, and $y = 3$, so that the solution is perfectly general, and equally applicable where *all* the roots of x and y are irrational.

This method should be interesting, moreover, because it considers the solution of cubic and biquadratic equations by direct means; whereas the modern tendency in engineering schools, and at the Naval Academy, has been rather to pass up the field of higher algebra, assuming that for all practical purposes, numerical approximation with a few refinements, is good enough for the solution of equations of the third degree or higher.

An effort has been made to show that only a very minimum of mathematical information need be retained in order

to accomplish the direct solution of such equations; and to this end, the complete development of each equation used has been shown.

$$x^2 + y = 7 \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$x + y^2 = 11 \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

From which we get,

$$x + (7 - x^2)^2 = 11$$

and,

$$x^4 - 14\,x^2 + x + 38 = 0 \quad . \quad . \quad . \quad . \quad (3)$$

Consider next the equation,

$$x = \sqrt{u} + \sqrt{v} + \sqrt{w} \quad . \quad . \quad . \quad . \quad (4)$$

Squaring,

$$x^2 = u + v + w + 2\,[\sqrt{u\,v} + \sqrt{u\,w} + \sqrt{v\,w}]$$

$$x^2 - (u + v + w) = 2\,[\sqrt{u\,v} + \sqrt{u\,w} + \sqrt{v\,w}]$$

Squaring again,

$$x^4 - 2\,x^2\,(u + v + w) + (u + v + w)^2$$

$$= 4\,(u\,v + u\,w + v\,w) + 8\,[\sqrt{u^2\,v\,w} + \sqrt{v^2\,u\,w} + \sqrt{w^2\,u\,v}]$$

$$= 4\,(u\,v + u\,w + v\,w) + 8\,\sqrt{u}\,\sqrt{v}\,\sqrt{w}(\sqrt{u} + \sqrt{v} + \sqrt{w})$$

and,

$$x^4 - 2\,x^2\,(u + v + w) - 8\,x\,\sqrt{u}\,\sqrt{v}\,\sqrt{w} +$$
$$(u + v + w)^2 - 4\,(u\,v + v\,w + u\,w) \quad . \quad . \quad . \quad (5)$$

If (5) is identical with (3), then by equating co-efficients, we have,

$$2\,(u + v + w) = 14 \qquad u + v + w = 7$$

$$-8\,\sqrt{u}\,\sqrt{v}\,\sqrt{w} = 1 \qquad \sqrt{u}\,\sqrt{v}\,\sqrt{w} = -\frac{1}{8}$$

$$(u + v + w)^2 - 4\,(u\,v + u\,w + v\,w) = 38$$

which gives,

$$(u\,v + v\,w + u\,w) = \frac{11}{4}$$

Consider also the equation,

$$(y - u)(y - v)(y - w) = 0 \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where u, v, and w are the roots of the equation, and,

$$y^3 - (u + v + w)y^2 + (u\,v + u\,w + v\,w)y - u\,v\,w = 0 \quad (7)$$

This being true, we may substitute for the coefficients, so that

$$y^3 - 7\,y^2 + \frac{11}{4}\,y - \frac{1}{64} = 0 \quad . \quad . \quad . \quad . \quad . \quad (8)$$

We can eliminate the squared term by the substitution,

$$y = Z + \frac{7}{3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and,

$$Z^3 + 7\,Z^2 + \frac{49}{3}\,Z + \frac{343}{27} - 7\left(Z^2 + \frac{14}{3}\,Z + \frac{49}{9}\right) +$$

$$\frac{11}{4}\left(Z + \frac{7}{3}\right) - \frac{1}{64} = 0$$

$$Z^3 - \frac{163}{12}\,Z - \frac{32843}{1728} = 0 \quad . \quad . \quad . \quad . \quad (10)$$

This equation has three real roots. It cannot therefore be solved by Cardan's method because it is impossible to extract the cube root of an imaginary number by algebraic means. We can solve it, however by comparing coefficients with those of the trigonometric identity below.

$$\sin^3 A - \frac{3}{4}\sin A + \frac{1}{4}\sin 3A = 0 \quad . \quad . \quad (11)$$

Letting,

$$Z = n \sin A \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

(10) becomes

$$n^3 \sin^3 A - \frac{163}{12}\,n \sin A - \frac{32843}{1728} = 0$$

or,

$$\sin^3 A - \frac{163}{12n^2}\sin A - \frac{32843}{1728n^2} = 0 \quad . \quad . \quad . \quad (13)$$

Now if (11) is identical with (13), we have, equating coefficients,

$$\frac{163}{12n^2} = \frac{3}{4}, \qquad n = \frac{\sqrt{163}}{3}$$

$$\frac{1}{4}\sin 3A = \frac{-32843}{1728n^3}, \qquad \sin 3A = \frac{-32843}{16(163)^{\frac{3}{2}}}$$

Evaluating,

32843				log	4.516	44282
16	log	1.204	11998			
163	log	2.212	18760			
163	½ log	1.106	09380			
		4.522	40138		4.522	40138
80°—31′—50″.14				sin	9.994	04144

$3A = -(80°—31′—50″.14) = 279°—28′—09″.86$

A_1 93°—09′—23″.29

A_2 213°—09′—23″.29

A_3 333°—09′—23″.29

Evaluating (12) for the three values of Z, we have

163	½ log	1.106	09380					
3	log	0.477	12125					
n	log	0.628	97255	0.628	97255	0.628	97255
A_1 93°—09′—23″.29								
	sin	9.999	34063					
Z_1 4.249259								
	log	0.628	31318					
A_2 213°—09′—23″.29				sin	9.737	92963		
Z_2 —2.327567				log	0.366	90218		
A_3 333°—09′—23″.29							sin	9.654 71118
Z_3—1.9216918							log	0.283 68373

From (9) we can now obtain u, v, and w.

	2.333333	2.333333	2.3333333
Z_1	4.249259		
u,	6.582592		
Z_2		−2.327567	
v,		0.005766	
Z_3			−1.9216918
w,			0.4116415

We have,

$$x = \sqrt{u} + \sqrt{v} + \sqrt{w} \quad . \quad . \quad . \quad . \quad (4)$$

From this it may be seen that there are eight possible combinations of signs.

+ + +	+ + −	+ − +	− + +
− − −	− − +	− + −	+ − −

But from (5) we know that

$$\sqrt{u}\ \sqrt{v}\ \sqrt{w} = -\frac{1}{8}$$

This reduces the number of possible combinations which satisfy the equation to four

− − −	+ + −	+ − +	− + +

These combinations give the following values of x,

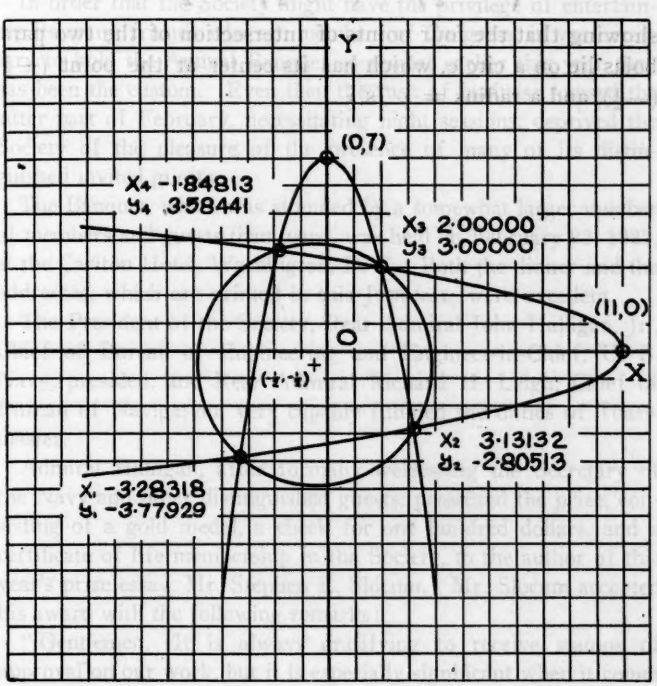
$\sqrt{u}$,	−2.565656	2.565656	2.565656	−2.565656
$\sqrt{v}$,	−0.641593	0.641593	−0.641593	0.641593
$\sqrt{w}$,	0.075934	−0.075934	0.075934	0.075934
	−3.283183	3.131315	1.999997	−1.848129
	x_1	x_2	x_3	x_4

Putting these values of x back in (1), we obtain the corresponding valves of y, so that we have the following pairs of answers,

x_1 —3.28318 x_2 3.13132 x_3 2.00000 x_4 —1.84813

y_1 —3.77929 y_2 —2.80513 y_3 3.00000 y_4 3.58441

The above solution has been carried through without any consideration of the geometric properties of the two parabolas



represented by the equations. The accompanying plot of the loci has been added, however, merely to show graphically the actual location of the roots.

It is interesting to note that by adding (1) and (2)

$$x^2 + y = 7 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$x + y^2 = 11 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

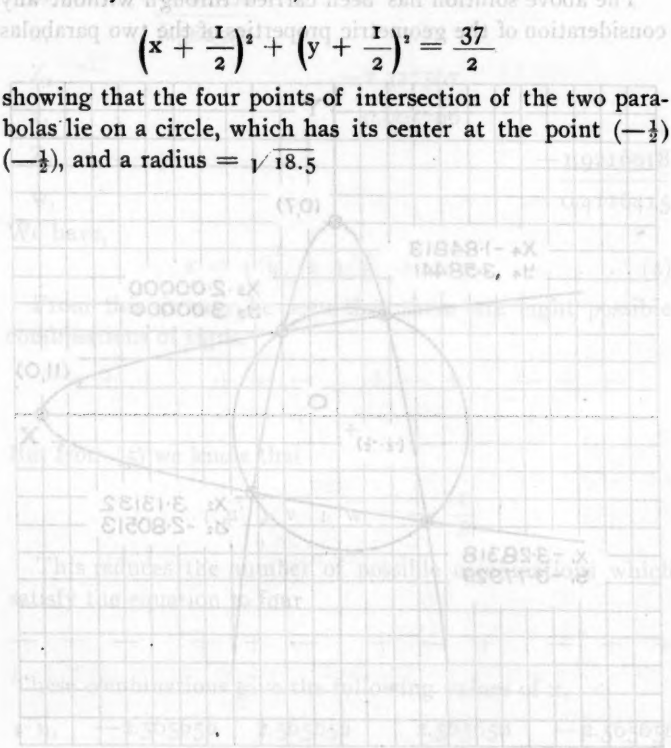
we get,

$$x^2 + x + y^2 + y = 18 \quad . \quad . \quad . \quad . \quad . \quad (14)$$

which may also be written,

$$\left(x + \frac{1}{2}\right)^2 + \left(y + \frac{1}{2}\right)^2 = \frac{37}{2}$$

showing that the four points of intersection of the two parabolas lie on a circle, which has its center at the point $(-\frac{1}{2})$ $(-\frac{1}{2})$, and a radius $= \sqrt{18.5}$



THE ANNUAL BANQUET.

In order that the Society might have the privilege of entertaining its many friends who are members of Congress, it was necessary to hold the Annual Banquet somewhat earlier this year than has been the custom. Even then the rush of business toward the latter part of February, necessitating night sessions, deprived the Society of the pleasure of the presence of many of its distinguished invited guests.

The Banquet, which was attended by a somewhat larger number of members and guests than usual, was held on February 23, 1927, at the Carlton Hotel, Washington, D. C. Both the dinner and the addresses, which are printed in this JOURNAL, were excellent.

The President of the Society, Rear Admiral John Halligan, Jr., Chief of Bureau of Engineering and Engineer-in-Chief, U. S. Navy, presided, and Rear Admiral Richard H. Leigh, Chief of Bureau of Navigation, very capably fulfilled the duties of Toastmaster.

Admiral Halligan, after formally welcoming the Secretary of the Navy and other distinguished guests, presented the prize, consisting of a gold medal, a check for one hundred dollars, and a certificate of life membership in the Society, to the author of this year's prize essay, Mr. Stephen E. Slocum. Mr. Slocum accepted his award with the following remarks:

"Gentlemen. It is always gratifying to receive stamps of approval on our work, but it is especially significant when it comes from such an eminent source as the American Society of Naval Engineers. I know of no other organization in which the membership measures up to such a uniformly high standard of ability and personality as does this Society. In expressing my deep appreciation of this award, I wish to assure you that I shall treasure this medal not only because it represents in a measure the success of my efforts to solve certain problems in hydrodynamics, but more especially because it is a token of your friendship and good will."

Admiral Halligan, in continuing his opening address told how, in 1888, the same year that Secretary Wilbur was graduated from the Naval Academy, the appropriation by Congress providing for the construction of seven cruisers, led to the formation and organization of our Society of Naval Engineers. Rear Admirals Baird, Griffin and Norton, who attended the Banquet were present at the first meeting of the Society, Rear Admiral Griffin being the first Secretary.

Before turning the meeting over to the Toastmaster, Admiral Halligan spoke of the high appreciation of Engineering in our service, and the co-operation from all other branches of the Navy which our naval engineers enjoy, which apparently is not the case in all other navies. The general interest in engineering is evidenced by the membership of our Society, including as it does, practically all corps as well as the line of the Navy, the Coast Guard and many members from civil life representing the many branches of engineering associated with the Navy.

Admiral Leigh, our Toastmaster, then took charge of the meeting and with very appropriate remarks in each case introduced the speakers, Assistant Secretary of the Navy for Aeronautics the Honorable Edward P. Warner, and Captain W. M. McFarland, and very successfully navigated the Banquet to its close.

The Honorable Mr. Updyke, Congressman from Indiana, was on the program to speak on "National Defense" but due to sudden illness he was unable to be present. Mr. Updyke is not only a splendid statesman, and a strong advocate of the Navy, but served overseas in the World War with the Seventy-fourth Company, Sixth Regiment, Second Division of the Marines, during which time he was wounded five times.

NOTES.

THE APPLICATION OF CIRCULATION THEORY TO A HYDRAULIC PROPELLER.

By Dr. H. Boerck.

"Jahrbuch der Schiffbautechnischen Gesellschaft" 1926, p. 306.*

Discussion by Dr. Bauer: After hearing the paper by Dr. Boerck and better after reading it one might come to the conclusion that a propeller may be designed with the aid of the circulation theory. I would like to warn not to attempt it.

It is natural that if a number of propellers are available which are very much alike, and which work under very similar conditions, then any one of a number of theories, and first of all the law of similarity, will give the desired results, provided these theories are based on the common laws of physics and provided further that any and all of the secondary influences actually existing in a ship in motion may be grossly neglected.

If the problems be however to design a propeller for a fast small ship and with very fine lines of the after body one day, and then for a freighter, and again for a passenger ship with a possible addition of a submarine propeller, and the designer would want to base his computations on theories that treat the whole problem in such summary manner, as for instance the wake theory, nothing but dire disappointments would follow such attempts.

The whole thing is not by any means so simple. The propeller problem is still a problem. It is probably even right to say that it will remain a problem, since the influences of variations in the ship's body on the propeller performance are so complicated, involving turbulence and eddies, which it may be truly said will set at naught all efforts of strict mathematical treatment. I wish to invite your attention only to the flow of water to the propeller. Who knows anything definitely accurate about this inflow? How fast is it? Where is it fastest? What direction has it? The experiment stations are attempting diligently to clear these points. Much credit is due to them for these efforts and such research must be encouraged in every way. I ask again: What effect have the lines of the after body? The arrangement of struts? How applicable is the law of similarity? One thing is certain—it does not apply here, since friction influences do not obey that law and they play a prominent role in the propeller problem. Hence a compromise is inevitable. When two similar phenomena are compared, as already mentioned, then one can make deductions from one to the other, but even for the most complicated curves can a straight line interpolation be used, when not too far away from the center of coordinates. Furthermore, how about cavitation? Where does it begin? When does it become noticeable? All these problems, gentlemen, have so far been neglected by these fundamental theories such as the circulation theory.

* Translated by E. C. Magdeburger.

I would also like to ask with regard to the assertion of the author, that "the torque within the usual limits is proportional to the square of the revolutions, the cube of diameters, the 1.5 power of pitch, the .8 power of blade width, first of the number of blades." Hence, if I make all these figures alike for the two propellers and have one with three blades and the other with four blades—does the latter take four-thirds of the torque of the first? That is a question. Undoubtedly this is not going to be so, but quite differently, as anybody that ever changed the number of blades on a propeller has learned.

How is it then when all else is equal and the pitch is changed? Is the torque proportional to the 1.5 power of pitch? I doubt it very much. But I must add expressly—a loop hole is necessary—that in propeller problems one must be very careful. I have not developed the proof of the formula given by the author and it is quite possible that I do not understand correctly this table of various powers above quoted.

To sum up, I must state again that in propeller computations one must be very careful, the moment that deductions about things quite far removed from the original scope of the general theory are made, these theories usually fail us quite completely. We can not depend on any theories, important though they undoubtedly are, in the study of the propeller, but must adhere strictly to the results of researches made in model basins, which discover and evaluate the most complicated influences, if false deductions on the performance of the propeller are to be eliminated.

THE AUTHOR'S REPLY: Dr. Bauer regrets that the circulation theory does not take into account the inflow of water into the propeller. The theory always gives the three characteristic velocities quite independently of any retardation, or acceleration of the water at the ship, as long as it is a free stream. Body lines must quite naturally be taken into account when designing propellers. In my paper I have stated that one must begin with a similar existing propeller, working under similar circumstances, if a propeller has to be designed based on these simple relations. Practice offers so many good examples of existing propellers, that such a basis for computations may always be found for any kind of condition.

In such manner a suitable propeller can always be designed, in fact better fitted propellers, than if they were based on results with model propellers, because with a model it is impossible to reproduce actual conditions, as they exist on a full-size ship. Besides all experiments with propeller models have the disadvantage that the problem is not handled analytically.

Circulation theory requires that the total loss be divided into its component parts, hence the treatment is distinctly not summary, but quite to the contrary, each individual factor affecting the efficiency of the propeller is studied independently, in order to obtain a propeller with the best possible total efficiency.

NOTES ON CONDENSER TUBES.

MATERIAL AND MANUFACTURE.

The Navy Specifications for condenser tubes of admiralty metal (70 Cu, 29 Zn, 1 Sn) call for a material which should give very satisfactory service provided that the design and construction of the condenser is suitable and the operating conditions are satisfactory.

In the manufacture of the tubes care should be taken that the maximum limits for impurities, which are iron—0.06, and lead—0.075, are not exceeded; that the surfaces are smooth and not scratched by the drawing, and that the

metal is properly annealed before each successive draw and after the final drawing process. The tube metal should be as homogeneous as possible. The design of the condensers should avoid direct impingement of exhaust steam on the tubes, should reduce the agitation and turbulence of the circulating water to a minimum. Careful operation is of course of great importance. In installation the tubes should not be over expanded or necked. There should be sufficient zincs in the water chests, and these should be kept in good condition and in close metallic contact with the shell. Tubes should be kept clean of heavy deposits such as mud, sand and vegetable matter, but the light gray or brownish scale in the tubes should not be removed as this is a decided help in preventing corrosion and pitting.

TUBE FAILURES.

Pitting which is such a common cause of tube failures is the result of dezincification, or eating away of the zinc. There have been many cases in the past where early failure from this cause could be traced to the use of muntz metal (60 Cu, 40 Zn) tubes, a material inferior to admirality brass for salt water use. Excessive trouble in certain ships is sometimes very mysterious, and when it cannot be traced to the ordinary causes, such unusual ones as the condensers being subjected to stray electrical circuits are looked for.

PROTECTIVE COATINGS.

A good many of the tube failures due to pitting can be definitely traced to the period in a ship's life between the time of launching and arrival in sea water. The ships are generally in fresh or brackish water of a polluted nature, and the thin scale of gray or brown nature, which is formed in sea water is not present to prevent the pitting which immediately sets in. To guard against this, experiments have been and are under way in this country by the American Society of Mechanical Engineers, the U. S. Naval Engineering Experiment Station, Annapolis, Md., and other bodies to develop satisfactory protective skins. The so-called "black skin" tubes of admiralty metal with the surfaces oxidized offer possibilities. Light artificial coatings of salt scale or films formed of sodium silicate appear to show decided promise to prevent this pitting. Investigation is being made to see whether such films cannot be formed on condenser tubes in operation, and it now appears probable that this can be readily done.

NEW TUBE MATERIALS.

Considerable work is being conducted both in Europe and in this country to develop new materials which will have greater resistance to corrosion and pitting than the brass tubes. Monel, nickel, nickel alloys, and some of the corrosion resisting sheets are being investigated. Suitable material must be found that is not only corrosion resisting but which can be manufactured in tube form at a reasonable price. In Great Britain the use of nickel alloy tubes in actual ship installations has been under test for several years. Two different compositions. are being used, with results so far indicative of excellent character, 80 per cent copper—20 per cent nickel and 70 per cent copper—30 per cent nickel. Except in special cases, these were installed with brass tube sheets and ferrules. The use of these materials has apparently eliminated the serious trouble previously encountered. In this country experiments of this nature have not been as advanced, but now that tubes of these various materials can be satisfactorily produced, extensive experiments in the American merchant marine and navy will probably result.

puted by the ordinary formulas of mechanics of materials are much lower than the stresses developed over many minute areas in the metal, perhaps computed stresses are only a small fraction of the actual localized stresses existing in structural and machine parts.

In recent years the X-ray spectroscope has given a picture of the atoms in a crystalline grain of metal, held together by forces whose nature is as yet a mystery, and arranged in some regular geometric pattern with a border region at grain boundaries having a more or less irregular atomic arrangement. The regular pattern of atoms which is repeated to make up a crystal is known as the space lattice of a metal, and from the viewpoint of the student of atomic structure elastic strain consists of a slight distortion of the space lattice, which distortion disappears when stress is released.

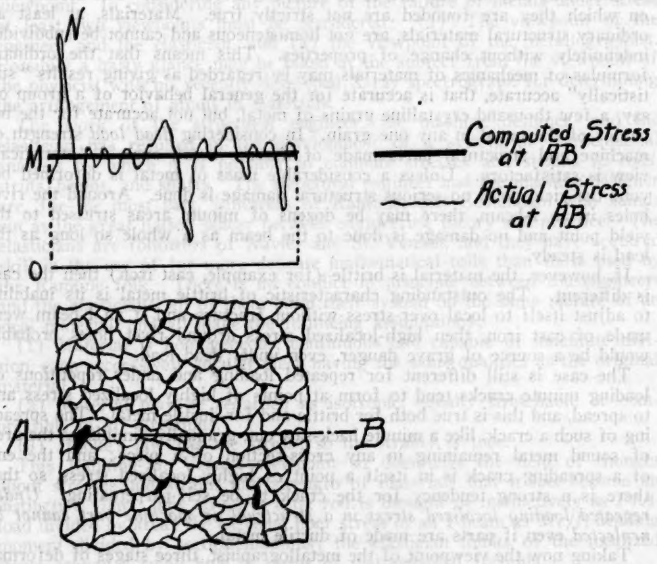


FIGURE 1.

As stress is increased in ductile material there comes about a state of affairs such that along certain planes of weakness in crystalline grains atomic bonds are broken. The divorced atoms slide over a few thousand other atoms, after which most of them find new partners and form new bonds with them and the remarkable thing is that the new bonds seem to be stronger than the old, after a brief period of restful adjustment. This action is known as "slip" and is shown under the metallographic microscope by a series of lines across a crystalline grain. Such lines are known as "slip lines" or "slip bands," and their appearance is shown in Figure 2.

Slip may be pictured as an action analogous to that shown by a pack of cards pressed together, face to back, and subjected to slightly oblique end-

wise pressure, such an arrangement as that shown in Figure 3*a*. Under a sufficiently heavy push the pack would take a position like that shown in Figure 3*b*. The end view of the pack is the end view of a stepped surface, and it is the edgewise view of these steps that shows the slip lines through the microscope. If the cards are slipped repeatedly the faces and the backs would become roughened and would offer increasing resistance to further slip.

To the metallographist a major significance of slip is the strengthening of planes of weakness within a crystalline grain. If slip could be brought about with no other effects than the exchange and strengthening of atomic bonds it would be an entirely beneficial process so far as strength is concerned. In some cases, *e.g.*, cold-drawn steel, the process is somewhat beneficial to the strength.

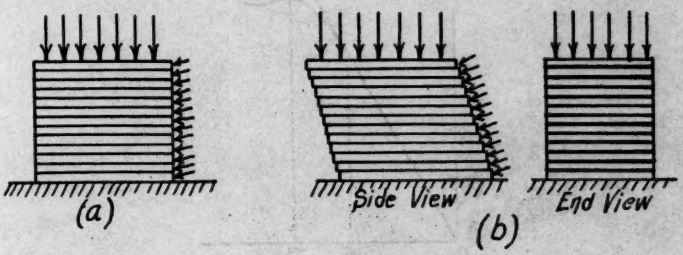


FIGURE 3.

However, it is not possible for the process of slip to go on without there being some locations where atomic bonds are broken and new bonds are *not* formed, that is minute, sub-microscopic cracks are developed. The earliest metallographic picture of the mechanism of fatigue failure of metals was a picture of cracks developing at slipping surfaces, growing to visibility under the microscope, and finally spreading to failure.* This is still a fairly satisfactory picture, although two other pictures of the origin and spread of cracks have been recently presented—pictures which do not picture cracks as necessarily originating at slipped surfaces. These pictures will be shown in succeeding paragraphs.

From the viewpoint of the engineering designer slip marks the elastic limit of metal. The commonly used definitions of elastic limit, proportional limit, and yield point are written in stress-strain nomenclature and the terms are not always used carefully. If elastic limit is considered from a metallographic viewpoint, the view may be less foggy. All the various kinds of elastic limits and yield points which are reported from laboratory tests of metals may be regarded as more or less delicate indicators of the beginning of appreciable slip in metal. At low stresses, probably under ordinary working stresses, slip occurs within a very few crystalline grains, but the number of grains so affected is so small that the total slip cannot be detected even by the use of a delicate extensometer. Under somewhat larger stress (*AE, Figure* 4) slip occurs within a larger number of crystalline grains

* Ewing and Humfrey, "Fracture of Metals under Repeated Alternations of Stress," Phil. Trans. Roy. Soc., A, 200, p. 241 (1903).

and on a carefully plotted stress-strain diagram, such as is shown in Figure 4, a slight deviation from the straight line relation can be detected, or if the load is released, a small permanent set remains. The value of stress for this first detectable irregularity in the stress-strain diagram is reported as proportional limit, or elastic limit. Its precise location depends to a marked degree on the delicacy of instruments used and on the precision of plotting of the diagram. When stress is increased to *By* (Figure 4), a general slip takes place throughout the metal under stress, affecting many grains, and stretch can be detected by direct measurement with dividers and a scale—some stretch occurs within the application of any additional stress, and in a testing machine test the beam "drops." This stage marks the yield point of the metal.

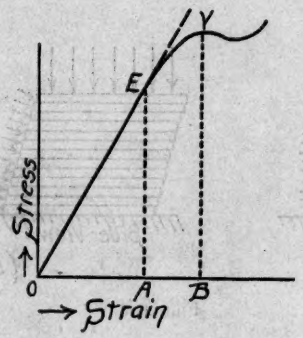


FIGURE 4.

If the strain on a metal is continually increased beyond the original strain where general slip takes place, actual fracture finally occurs. In the case of brittle materials, such fracture occurs before slip becomes widespread enough to show a well-marked yield point. Under a single loading in tension the final fracture of a metal, either a ductile metal or a brittle metal, appears to take place simultaneously over the whole section of a piece of metal, that section seems to act like the famous "one-horse shay" which went to pieces "all at once and nothing first, just as bubbles do when they burst." A careful study of the bursting of bubbles and the failure of tension test-pieces shows that in both cases the actual fracture is progressive, not instantaneous. If the fracture of a tension test-piece be examined, there usually can be found evidence that the failure began at some definite region and spread across the piece, although its spread was rapid. In some ductile metals fracture can be seen to progress across the test specimen. Fracture under a single loading is a very rapid progressive fracture. It may be safely stated that no experimenter has ever loaded a test-piece of metal so carefully and so accurately that all the atomic bonds on a cross-section were broken at the same instant.

Under repeated loading a stress well below the ultimate tensile strength will start a fracture in metal which finally spreads to failure of the entire cross-section of a piece. This spread under repeated loading is very much slower than the spread under a single increasing load. Thousands or even

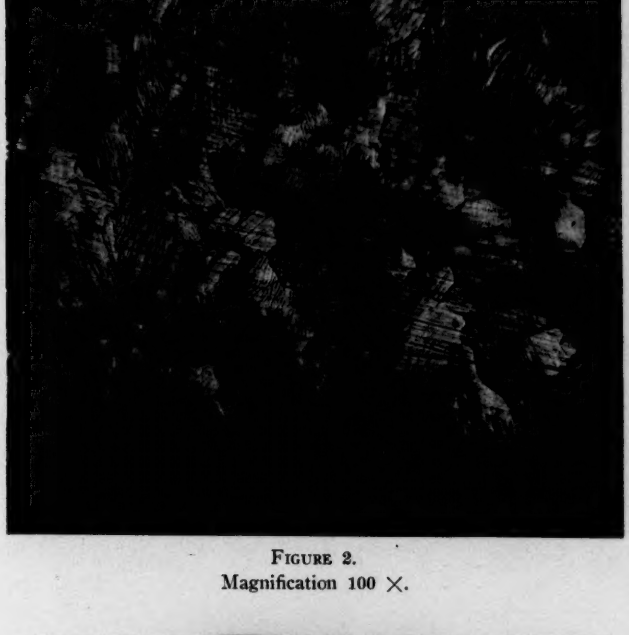


FIGURE 2.
Magnification 100 ×.

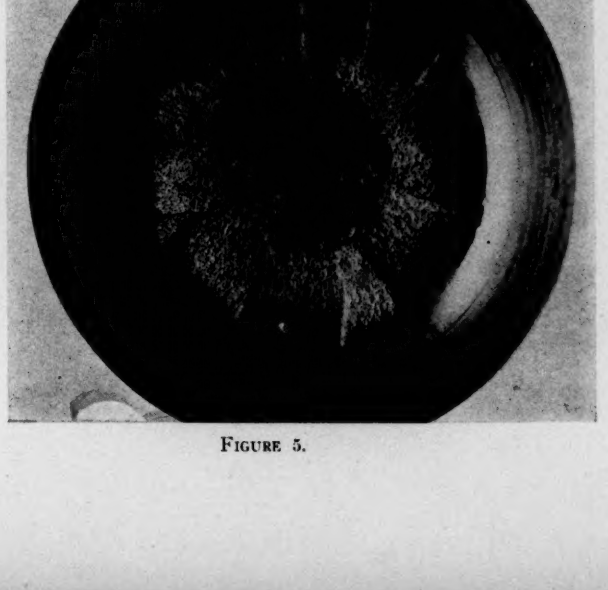
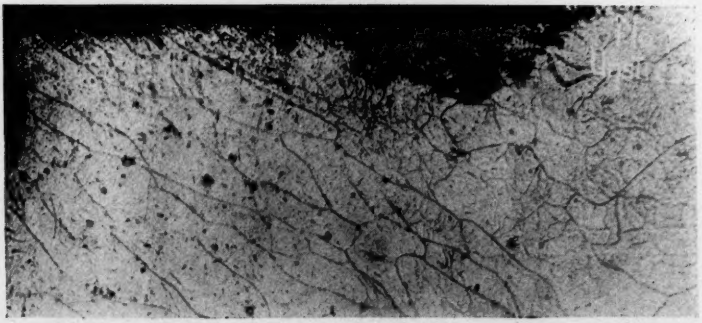
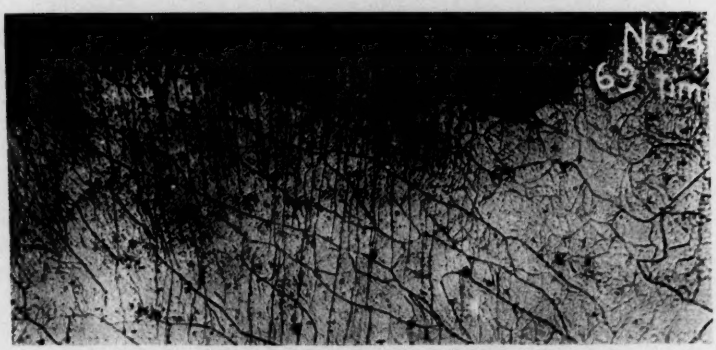


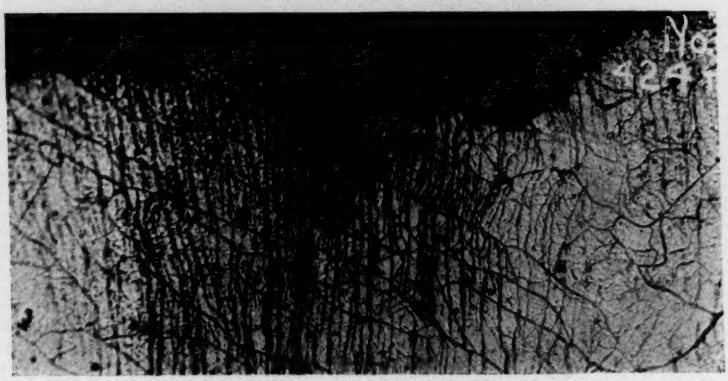
FIGURE 5.



(a) unstressed.



(b) After 69 cycles of severe flexure.



(c) After 424 cycles of severe flexure.

FIGURE 6.

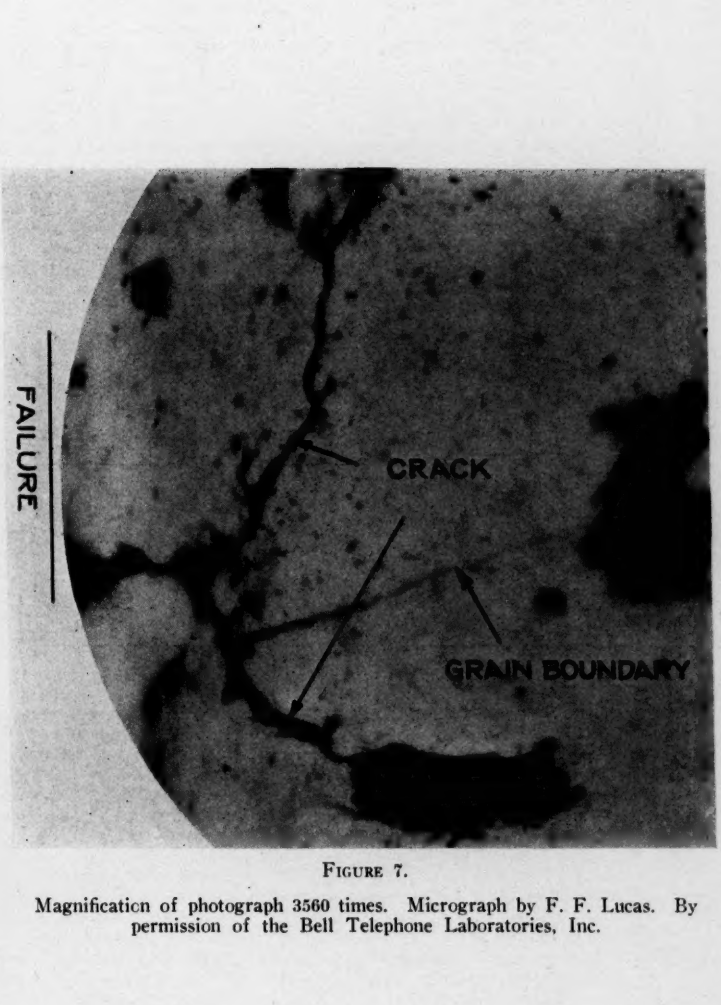


FIGURE 7.

Magnification of photograph 3560 times. Micrograph by F. F. Lucas. By permission of the Bell Telephone Laboratories, Inc.

FIGURE 10.

Crack in ferrite-pearlite steel. The crack was filled with a deposit of copper before the piece was polished and hence appears light colored. Note how the crack skirts the strong pearlite grains, and the rounded ends of the crack. Magnification about 350 times. Micrograph furnished by H. A. Schwartz of the National Malleable and Steel Castings Company.

millions of cycles of stress may be required to develop the final failure of a machine part. Not infrequently the spreading crack can be detected before it has progressed to failure and a disaster averted. This repeated stress fracture spreads slowly like a minute hack-saw cut, but its rate of progress is accelerated, and just before fracture, it is almost as rapid as is the spread of fracture under a single increasing load. In fact, a typical fatigue failure usually shows two distinct zones: (1) A smooth surface where the crack has spread slowly and the walls of the crack are battered smooth by repeated opening and closing, (2) a rough "crystalline" surface indicating the very much more sudden fracture of the core of the piece.

Figure 5 shows the fracture of a rotating-beam test specimen subjected to cycles of repeated flexure. Fracture started at the outer circumference, and a crack gradually spread inward. The walls of this crack were continually shoved against each other as the crack opened and closed under successive cycles of stress. The walls of this part of the crack are worn smooth and occasionally little longitudinal breaks occurred, leaving steps in the surface which roughly resemble ripple marks left on sand by flowing water. When the crack has spread to this inner circle shown in Figure 5, the failure of the remaining metal progressed so rapidly that a rough "crystalline" surface was left, such a surface as is found when a shaft with a sharp shoulder is fractured under a single very heavy load.

Definite knowledge as to the nature of fatigue cracks in their initial stages is entirely lacking. Attempts to use the microscope to detect fatigue cracks in their very early stages have not met with much success. If, however, a specimen is notched so as to localize fractures at some definite cross-section, such fatigue cracks in large crystalline grains of metal can be detected quite early in the "life" of the piece, and they can be seen to multiply and to lengthen under successive cycles of stress. Figure 6 shows three views of a specimen of armco iron subjected to violent reversals of flexure and the multiplication and lengthening of cracks is evident. It is, however, exceedingly difficult to detect fatigue cracks in small-crystalled metal, and it is exceedingly tedious to hunt for microscopic cracks over any considerable area of surface of metal. Figure 7 shows a very small fatigue crack in armco iron taken at the magnification of 3560 times. It is to be noted that the crack goes straight across grain boundaries and goes out of its direct course to take in a non-metallic inclusion. In spite of its difficulties the microscopic study of fatigue cracks in metals offers a very promising field for the investigator. The question of initial stress in metals and of whether such regions of stress are sources of fatigue cracks also needs investigation. Some locations where there are no cracks may be under high internal stress—stress so high that but little additional stress is necessary to start a crack.

In structural members and machine parts, it is sometimes possible to detect fatigue cracks before they have spread to fracture. Some railroads and street railways make a practice of inspecting axles of cars and locomotives at regular intervals to see if such small cracks can be detected. When this is carefully done there have been very few disasters due to fatigue fractures in axles. In some experiments now in progress at the University of Illinois, it has been found possible to detect cracks in specimens 1⅛ inches in diameter, when about one-half the "life" of the specimen has passed, unless the applied stress is very high. These specimens were cut from car axles.

The whole question of fracture in metals brings up the relation of theoretical cohesion and strength. From a determination of the latent heat of fusion and the latent heat of vaporization, physicists have computed the theoretical cohesion of atoms for many metals, and if cohesion in solids is of the same order of magnitude as cohesion in melting solids and vaporizing

liquids, then the tensile strength of most metals should be from fifteen to twenty times as great as it is found to be in tests.

The most obvious explanation of this great difference is that in our ordinary metals the system of atomic bonds is far from perfect. It has already been noted that the elastician's picture of continuous, homogeneous material is not true for ordinary metals. If we look at metals from the viewpoint of the metallographist, we might draw a rough picture of metals by considering them as continuous but not homogeneous. If we think of imperfect bonds between atoms, minute cracks, severe internal stresses which when slightly increased will produce cracks, we may draw a picture from the engineer's viewpoint of metal which is homogeneous but not continuous. Neither of these pictures can claim to be complete. The metallographist's picture does not lend itself to mathematical computations of strength. The engineer's picture of metal, which may be regarded as homogeneous but which has in it many small holes or many irregularities of outline, *does* lend itself to such computation. The mathematical theory of elasticity and the mechanical means which can be used to solve some of its more complex equations can be employed to determine approximately the effect of these supposed discontinuities. With this apologia for using a method of analysis which is admittedly based on an incomplete picture, but which is believed to be useful, the writer would like to develop in some detail a picture of the behavior of metal under stress.

In developing this picture, the writer is following outlines given by two physicists, one the British physicist, A. A. Griffith* and the other the Russian

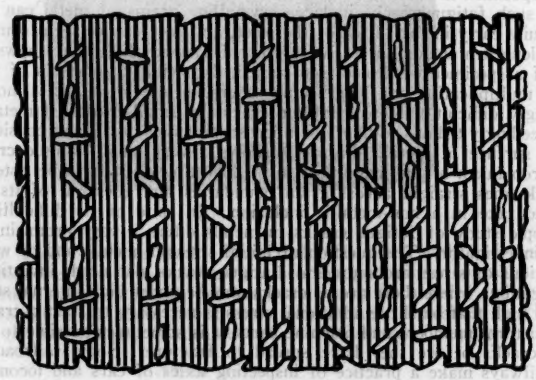


FIGURE 8.

physicist, Joffé. These two men present pictures which are not contradictions, but they do emphasize two different sources of irregularity. In Griffith's picture the possibility of minute discontinuities throughout the metal are emphasized. Figure 8 is a cartoon of the Griffith idea. Metal is pictured as having in it a multitude of minute cracks, cracks say, 0.0002 inch long and a few hundred atoms wide. Such cracks could not be detected by

* "Phenomena of Rupture and Flow in Solids," Phil. Trans. Roy Soc., A, 221, p. 163 (1920).

any present-day microscope. If such cracks exist, they must be very numerous and must be scattered throughout the metal, else the metal could not be produced with such dependable physical properties as is found to be the case. These minute cracks weaken the metal in two ways: (1) By diminishing the area of the cross-section, and (2) by causing very high localized stress at the ends of the crack. Again attention must be called to the limitations of the theory of elasticity and to the improbability that these formulas would apply with any high degree of accuracy to such small areas of metal as are involved in considering these cracks; however, the general qualitative conclusions of the theory of elasticity may be expected to furnish a useful guide for estimating the general effect of such cracks. From such general conclusions it would seem that if Figure 9 represents such a small crack the

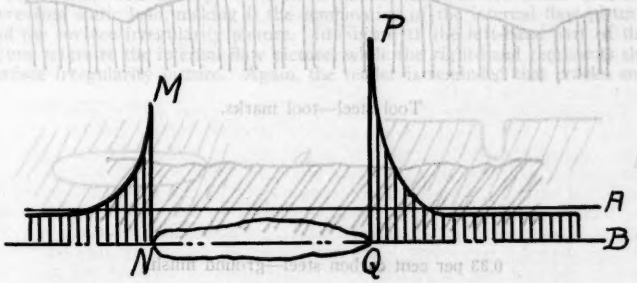


FIGURE 9.

AB, mean stress in region near crack. *NM-PQ*, localized stress at ends of crack.

stress intensification at the ends is a function of the direction of the long axis of the crack with respect to the direction of the stress and a function of the sharpness of curvature at the end of the crack. The reader is reminded that all cracks have rounded ends, and is referred to Figure 10 for an illustration of this.

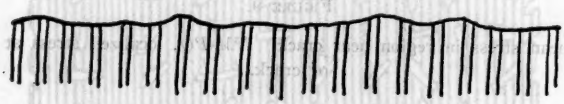
The picture given by Joffé emphasizes the irregularities at the surface of the metal as a source of highly localized stress. All metallic surfaces have minute hills and hollows which are, however, large when compared with atomic dimensions, and these hills and hollows are the sources of stress intensifications, just as are the internal cracks pictured by Griffith. Figure 11, which shows actual surface irregularities magnified three hundred and sixty times, may be used as the Joffé picture. Evidently there will be stress concentration at the bottom of the minute notches in the surface of the metal and the magnitude of that stress concentration at the root of any notch depends upon depth of the notch and the sharpness of curvature at the root of the notch.

Griffith has obtained some evidence for his theory from tests of glass in which cracks have been produced artificially and also from tests of glass drawn out into fine threads. Joffé has obtained some evidence for his picture from tests of a single crystal of a salt ground to spherical shape and subjected without failure to extreme stress at the middle by the process of

0.13 per cent carbon steel—tool marks.

Tool steel—tool marks.

0.33 per cent carbon steel—ground finish.



Tool steel—ground finish.

0.33 per cent carbon steel—reamer finish.

Magnification 360 times. Traced from micrographs of gelatin casts obtained by W. Norman Thomas; See Reports and Memoranda No. 860 of the British Aeronautical Research Committee. By permission of the Controller of His Britannic Majesty's Stationery Office.

FIGURE 11.

immersing it in liquid air and, when it has thoroughly cooled, plunging it into molten lead. Griffith has noted the possibility of stress concentrations at the surface of metals and Joffé admits the possibility of stress concentration between crystalline grains of metal. As noted above, the two pictures are supplementary and not contradictory.

As noted in a previous paragraph, there was an earlier picture of the mechanism of fatigue failure in which cracks were shown as having their origin in planes of slip. This picture is not contradictory to the internal flaw picture given above. Whether the cracks develop after slip or are always present in metal once present, the phenomena of stress intensification and of spread may be imagined as proceeding, as shown in the internal flaw picture.

The writer will attempt to draw a picture of fracture under steadily increasing static load, making it the combination of the internal flaw picture and the surface irregularity picture. In Figure 12 the left-hand part of the figure refers to the internal flaw picture, while the right-hand represents the surface irregularity picture. Again, the reader is reminded that cracks and

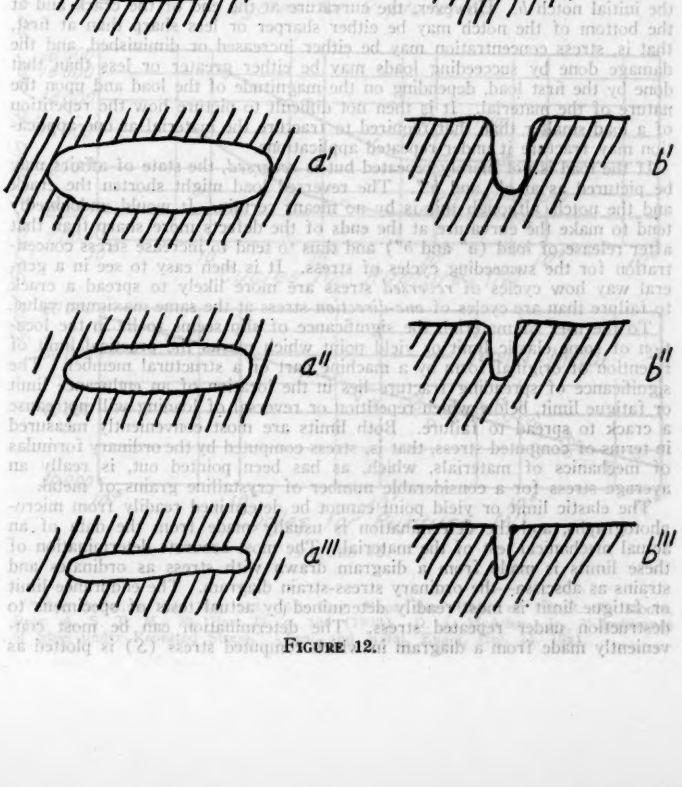


FIGURE 12.

notches have roughly rounded ends. (Figure 10.) Under increasing load at least three things happen: (1) The hole, shown at a, and the notch, shown at b, spread to the condition shown at a' and b', respectively; (2) the curvature of the ends of the crack and the curvature at the bottom of the notch become less sharp, causing diminution of stress concentration and benefit to the metal; (3) slip, as well as fracture, occurs at the ends of the crack and at the bottom of the notch, probably causing diminution of internal stress, and tending to increase the strength of the material. Under a moderate increase of loading, equilibrium is reached and the damage caused by the crack (or the notch) spreading slightly, causing a reduced cross-section, is balanced by the diminished stress concentration at the ends of the crack or the notch, and furthermore by the tendency to strengthen the material at the points of highest stress. Under a sufficiently great increase of load the spread of the defects overbalances the beneficial effects of lowered stress concentration and failure occurs, and its final progress is very rapid. If the metal has low ductility the strengthening effect and the reduction of internal stress due to slip is slight, since brittle material slips very little before fracture.

To make this picture fit the case of repeated stress a third part must be added. If a load is applied not sufficient to cause failure and that load is then released, the resulting state of affairs may be pictured as at a'' and b''. The crack a'' is longer than the initial crack a and the notch b'' is deeper than the initial notch b. However, the curvature at the end of the crack and at the bottom of the notch may be either sharper or less sharp than at first, that is, stress concentration may be either increased or diminished, and the damage done by succeeding loads may be either greater or less than that done by the first load, depending on the magnitude of the load and upon the nature of the material. It is then not difficult to picture how the repetition of a load smaller than that required to fracture the material at one application may fracture it under repeated applications.

If the load is not merely repeated but is *reversed*, the state of affairs may be pictured as at a''' and b'''. The reversed load might shorten the crack and the notch, although this is by no means certain. It would undoubtedly tend to make the curvature at the ends of the defects more sharp than that after release of load (a'' and b'') and thus to tend to increase stress concentration for the succeeding cycles of stress. It is then easy to see in a general way how cycles of *reversed* stress are more likely to spread a crack to failure than are cycles of *one-direction* stress at the same maximum value.

To the user of material, the significance of slip seems to lie in the location of some elastic limit or yield point which marks the practical limit of retention of original form by a machine part or a structural member. The significance of spreading fracture lies in the location of an endurance limit or fatigue limit, below which repetition or reversal of loading will not cause a crack to spread to failure. Both limits are most conveniently measured in terms of computed stress, that is, stress computed by the ordinary formulas of mechanics of materials, which, as has been pointed out, is really an average stress for a considerable number of crystalline grains of metal.

The elastic limit or yield point cannot be determined readily from microphotographs, and the determination is usually made from the data of an actual mechanical test of the material. The most accurate determination of these limits is made from a diagram drawn with stress as ordinates and strains as abscissa—the ordinary stress-strain diagram. The endurance limit or fatigue limit is most readily determined by actual tests of specimens to destruction under repeated stress. The determination can be most conveniently made from a diagram in which computed stress (S) is plotted as

ordinates and number of cycles of stress required to cause fracture (N) is plotted as abscissa. Such diagrams are called stress-cycle diagrams or S-N diagrams. The endurance limit is determined when this S-N diagram becomes horizontal. Methods of making endurance tests have been described so frequently that it is not necessary to give their detail here.* It is usually convenient to plot the values of numbers of repetitions of stress required to cause fracture to a logarithmic scale, and in some laboratories stress is also

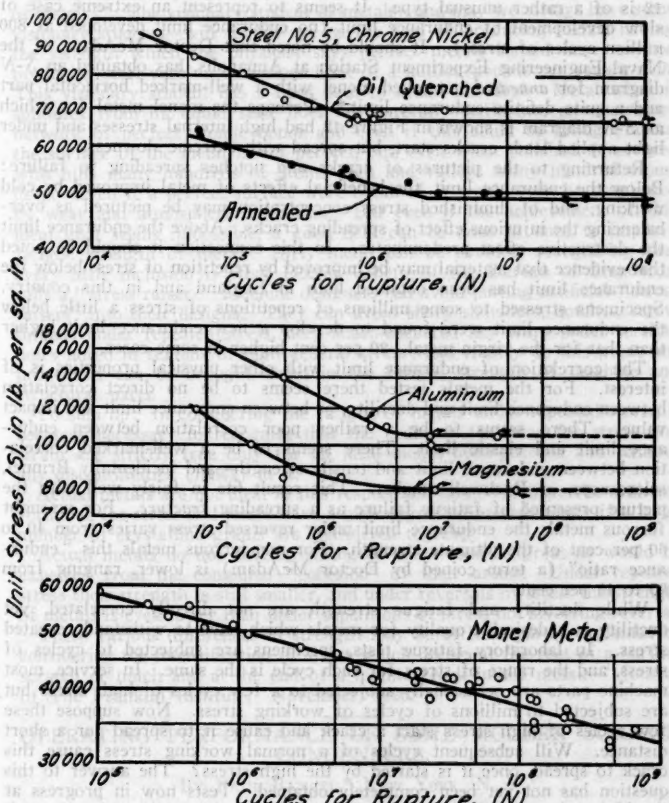


FIGURE 13.

* See Bulletins 124, 136, 142, 152, Eng. Expt. Sta. Univ. of Ill. R. R. Moore, Proc. A. S. T. M., 23, Part II, p. 106 (1923) D. J. McAdam, Jr., "Endurance of Steel under Repeated Stress," Chem. and Metal. Eng'g, Dec. 14, 1921.

plotted to logarithmic scale. Figure 13 shows typical *S-N* diagrams. The plotted points marked with arrows indicate specimens not broken.

From an examination of many of these *S-N*. diagrams the writer finds that, in general, they approach a horizontal direction as the number of cycles of stress is increased. The *S-N* diagrams obtained from various metals extend to as high as a thousand million cycles of stress and seem to give good evidence that endurance or "life" of metal will be indefinite if the computed stress is less than that for which the curve for a metal becomes horizontal. The *S-N* diagram for hot-rolled monel metal shown in Figure 12 is of a rather unusual type. It seems to represent an extreme case of slow development of endurance limit (no endurance limit developed at 800 million cycles of stress). It should be noted that Doctor McAdam, of the Naval Engineering Experiment Station at Annapolis, has obtained an *S-N* diagram for *annealed* hot-rolled monel with a well-marked horizontal part and a quite definite endurance limit.* Perhaps the monel metal for which an *S-N* diagram is shown in Figure 12 had high internal stresses and under light applied loads cracks start, but spread with extreme slowness.

Returning to the pictures of cracks and notches spreading to failure: Below the endurance limit, the beneficial effects of metal improved by cold working, and of diminished stress concentration may be pictured as overbalancing the injurious effect of spreading cracks. Above the endurance limit the destructive effect predominates. In this connection it should be noted that evidence that material may be improved by repetition of stress below the endurance limit has been obtained both in England and in this country. Specimens stressed to some millions of repetitions of stress a little below the endurance limit were found to develop a new endurance limit higher than that for the virgin metal—20 per cent higher in some cases.

The correlation of endurance limit with other physical properties is of interest. For the metals tested there seems to be no direct correlation between endurance limit and ductility, or between endurance limit and impact value. There seems to be a rather poor correlation between endurance limit and elastic limit. There seems to be a well-marked correlation between endurance limit and tensile strength—and incidentally Brinnel, scleroscope, or Rockwell number. This result fits in fairly well with the picture presented of fatigue failure as a spreading *fracture*. For wrought ferrous metals the endurance limit under reversed stress varies from 40 to 60 per cent of the ultimate strength. For non-ferrous metals this "endurance ratio" (a term coined by Doctor McAdam) is lower, ranging from 25 to 45 per cent.

While ductility and fatigue strength are not directly correlated, yet ductility is a desirable quality for metals which have to withstand repeated stress. In laboratory fatigue tests, specimens are subjected to cycles of stress, and the range of stress for each cycle is the same. In service, most machine parts are occasionally subjected to a few cycles of high stress but are subjected to millions of cycles of working stress. Now suppose these few cycles of high stress start a crack and cause it to spread for a short distance. Will subsequent cycles of a normal working stress cause this crack to spread, once it is started by the high stress? The answer to this question has not yet been completely obtained. Tests now in progress at the University of Illinois on car-axle steel have shown that once a crack starts it may spread under cycles of stress at least 35 per cent lower than the original endurance limit of the metal. It seems highly probable that the tendency for a crack to spread, once it is started, depends in some measure

* Such a diagram for annealed monel has very recently been obtained at Illinois.

on the ductility of the metal. Tests on specimens with sharp notches or holes which artificially cause high localized stress indicate in a general way that such artificial defects do more damage in the case of brittle materials than in the case of ductile materials, although test results show that there are evidently other factors concerned as well as ductility. Ductility in metals then may be looked on as furnishing some protection against destructive damage by occasional over-stress, although it is not a direct factor in increasing fatigue strength of metal.

The question may be asked, "What can be done to prevent fatigue failure of metals?" The visionary student of metals—and may we always have some visionary students—will fix his eye on the goal of producing metals having the strength which the physicist says perfect metals should have, say on producing steel with an endurance limit of a million pounds per square inch. This visionary student has a hard path to attain his ideal. Metal when it solidifies from its liquid state does not naturally settle into an unstressed network of perfect atomic bonds and even if this could be brought about the surface of the metal is not perfect, but, even when polished, is covered with hills many hundreds of atoms high and valleys many hundred atoms deep. Even if a perfect surface were obtained, it would soon be roughened by wear and atmospheric corrosion. But even if the achievement of this ideal goal seems impossible, a good deal can be done at once to improve fatigue strength of metal. Dirty metal can be avoided, recognizing that every little inclusion is what Doctor Gillett, of the Bureau of Standards, calls a "stress raiser." Machine designers can avoid putting artificial "stress raisers" into their designs. They can avoid sharp shoulders, notches, screw threads under repeated stress, deep sharp-cornered key ways under torsion, holes bored in regions of high general stress; and if they do this great improvement can be brought about in the fatigue strength of machine and structural parts.

To sum up: In considering the fatigue of metals under repeated stress, it becomes necessary to recognize that the ordinary formulas for stress and strain are strictly true only for an ideal metal, homogeneous and without any discontinuities (flaws) in it.

Actual metals are not ideal in this respect and hence for them the ordinary formulas for stress and strain have a high degree of precision only if a large number of crystalline grains are considered together.

Actual metals do not develop strengths under load so great as would be expected from the computed "cohesion" of their atoms. Under repeated stress their strength is still smaller, and under reversals of stress the strength of metals is even lower than under repetitions of stress not involving reversal.

These various differences in strength may be at least partly explained by considering the intensification of stress which occurs at the root of a surface scratch or notch and at the end of a minute crack in the metal.—"Journal of The Franklin Institute," November, 1926.

OPERATING PERFORMANCE OF SOME MODERN SURFACE CONDENSERS.

By Paul Bancel,* New York, N. Y.

By means of operation records, this paper shows (1) that a modern condenser requires about half the cooling surface of the conventional type; (2) that there is no essential difference between operating and test performance;

* Ingersoll-Rand Company. Mem. A. S. M. E.

Contributed by the Power Division and presented at the Annual Meeting, New York, December 6 to 9, 1926, of The American Society of Mechanical Engineers. All papers are subject to revision.

(3) that the results obtained are based on fundamental improvements in design which produce high efficiency in the utilization of the surface with small expenditure of power; and (4) that the change from a two-pass to a single-pass hydraulic circuit is the result of improved efficiency and not the cause.

When the Ingersoll-Rand condenser was introduced, it was claimed that the usual cooling surface could be greatly reduced; that is, 10,000 square feet would do the work of 20,000 square feet and 25,000 square feet the work of 50,000 square feet with equal auxiliary power in both cases. So radical a claim was of course immediately challenged. With one-half the surface, the heat-transfer coefficient would naturally be doubled, and criticism simmered down largely to the foolhardiness, or worse, of using so high a coefficient in the face of all the experience and practice of past years. It was argued that condensers of conventional design were not the crude machines of prewar days, but were, in fact, already highly efficient, as demonstrated by tests showing coefficients of 500, 600, and 700 as against those of 300 to 500 used when calculating the cooling surface. Therefore, using test results as a basis of size was sacrificing a very essential and large margin of safety.

One test of a 50,000-square-foot condenser at the Waterside Station of the New York Edison Company was frequently cited a few years ago (see 1924 N. E. L. A. Condenser Report, p. 43). A coefficient of 600 was shown under conditions of cold water and high vacuum where considerably lower heat transfer was expected than in summer, and 300 to 400 would have been considered excellent. Since that time the author's company has installed a unit in the same plant, and the same work is being performed with 60 per cent of the surface, with the same or less water, and with considerably less friction which is to be credited to the performance. While the previous unit had shown a coefficient of 600, the Ingersoll-Rand unit has never shown a 1000 or 1200 coefficient but, nevertheless, produces the same operating results year in and year out.

It is apparent, therefore, that many tests and fragmentary readings on condensers are of little or no value. As with a boiler plant, the real test of condenser performance is that obtained in continuous operation. Results of this kind are presented in the following pages and similar operating results should be used for comparison.

Comparisons should also be made with the latest single-pass condensers of other types. Recently it has been argued that the performance of any type of condenser can be improved by making it single-pass instead of two-pass. Many such condensers have been put into service within the last year or two which are of at least as recent design as those covered in this paper, and a comparison of complete operating results would be of great interest.

Table 1 gives details of an Ingersoll-Rand condenser of 9000 square feet surface carrying loads as high as 16,000 kilowatts and condensing 168,000 pounds of steam per hour, and also of other units referred to later. Figure 1 shows fifty days' performance in detail. Figure 2 shows a straight-line chart of the performance of this condenser with one set of points covering 1925 and the other 1926.

Calculations from the figures will show heat-transfer coefficients over 800, as would be expected. Figure 2 shows that the performance of the summer of 1925 was repeated in 1926. The loads approach 20 pounds per square foot, which is believed to be a record for a land condenser with circulating water at 80 to 90 degrees F. in Summer. The straight-line chart is used in Figure 3 and elsewhere because it is a convenient way of eliminating erratic values and has the advantage that the line drawn is made with a straight edge. Appendix No. 1 explains the straight-line law.

TABLE 1 DATA ON VARIOUS CONDENSERS

Location	Date of installation	Surface Sq. ft.	No. passes	Tubes				Circulation			Steam jets Operating steam, lb. per hr.
				No.	Size, in. O.D.	Gage	Length ft.	Water quantity, gal. per min.	Velocity ft. per sec.	Friction, ft.	Nature of circulating water
Broad River Power Co.	Parr, S. C.	Oct., 1925	9000	Single	2300	1/4	17	20	22,000	9.6	14.5	Broad River, fresh, muddy. Summer temperature 80° to 90°	1000
Broad River Power Co.	Parr, S. C.	Aug., 1926	25,000	Single	5800	1/4	17	22	45,000	7 8	12	Broad River, fresh, muddy. Summer temperature 80° to 90°	1000
N. Y. Edison Co. Waterside Station	Waterside No. 1, New York City	Oct., 1924	30,000	Single	6000	1/8	18	22	70,000 (rated) (55,000 to 60,000,) (operation in 1925)	7.9 6.5 (average)	10 7	Salt water, sewage contamination, New York, East River, summer temperature 70° to 75°	1800
City of Pasadena	Pasadena, California	May, 1924	14,100	Single	3582	1/4	18	20	22,000	5.9	7	Cooling tower, algae and slime form in tower and tubes. Summer temperature 90° to 100° maximum, 107°	1500

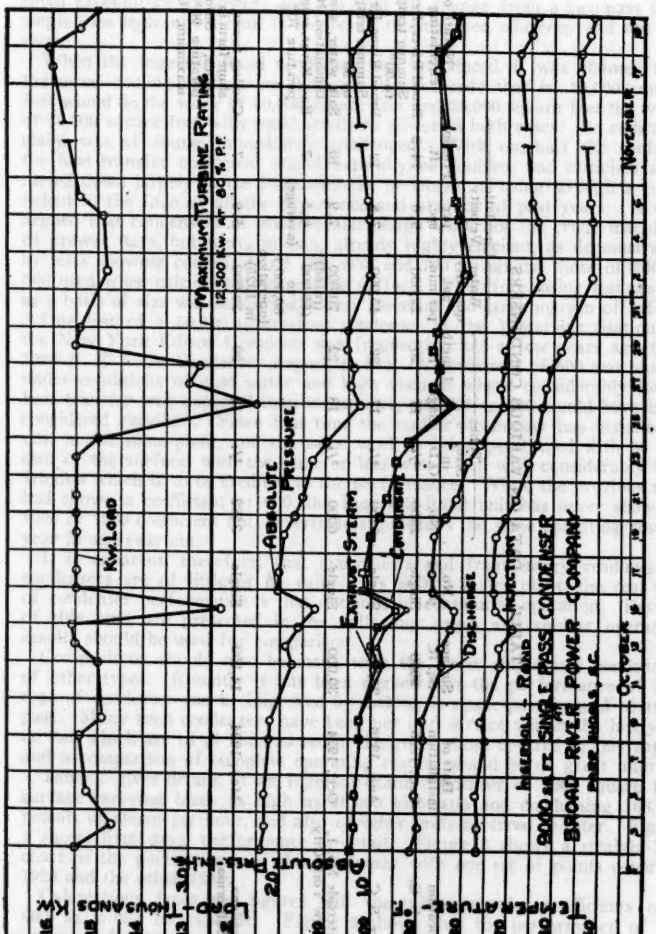


FIGURE 1.—CONTINUOUS GRAPHIC RECORD OF THE PERFORMANCE, DURING THE FIRST ONE AND ONE-HALF MONTHS, OF THE 9000-SQUARE-FOOT CONDENSER AT THE BROAD RIVER POWER COMPANY, PARR, S. C.

Figure 3 shows the performance of a 25,000-square-foot unit in the same plant serving a 38,000-kilowatt (maximum rating) Westinghouse turbine. This chart is also furnished by the W. S. Barstow Management Association.

Figure 4 shows the straight-line law based on a number of these readings as given. At 240,000 pounds load the difference between exhaust steam and water temperature is 17 degrees; at 300,000 pounds load it is 21.2 degrees or a vacuum of 28 inches with 80-degrees water. Comparison should be made with the monthly performance of 50,000-square-foot units installed for similar service within recent years, including load, water quantity, condenser friction, and air-pump power or steam-jet quantity.

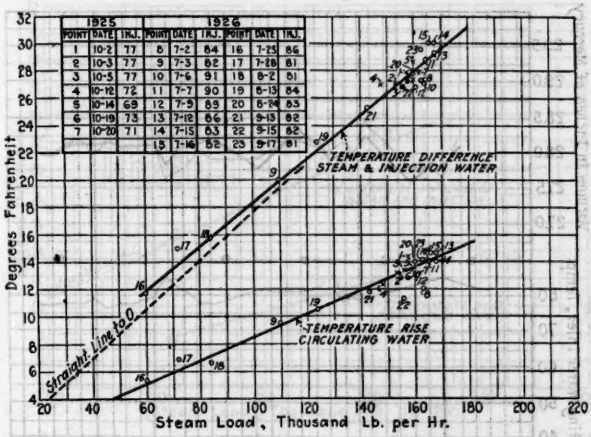


FIGURE 2.—PERFORMANCE IN SUMMERS OF 1925 AND 1926 OF THE INGERSOLL-RAND 9000-SQUARE-FOOT CONDENSER AT THE BROAD RIVER POWER COMPANY PLOTTED TO THE STRAIGHT-LINE LAW.

(The chart shows the difference between steam temperature and injection temperature; also the temperature rise of the circulating water. The table gives the actual injection temperatures. It will be noted that there is no difference in the performance in 1926 as compared to 1925.)

An interesting feature of this condenser is that it is hung entirely from the turbine. There are no spring supports. A second feature is the small basement height—22 feet from floor to floor.

Figure 5 shows about 50 days' operation and test results of a 30,000-square-foot condenser under a 35,000-kilowatt non-bleeding turbine at Waterside Station, New York Edison Company. The tests were based on 70,000 gallons per minute, the rated water, and are corrected to average operating water quantity by the usual method—lines A′ and B′. Due to local conditions the pumps (turbine-driven) were operated at reduced speed, and the water quantity averaged 55,000-60,000 gallons per minute as shown by the temperature rise.

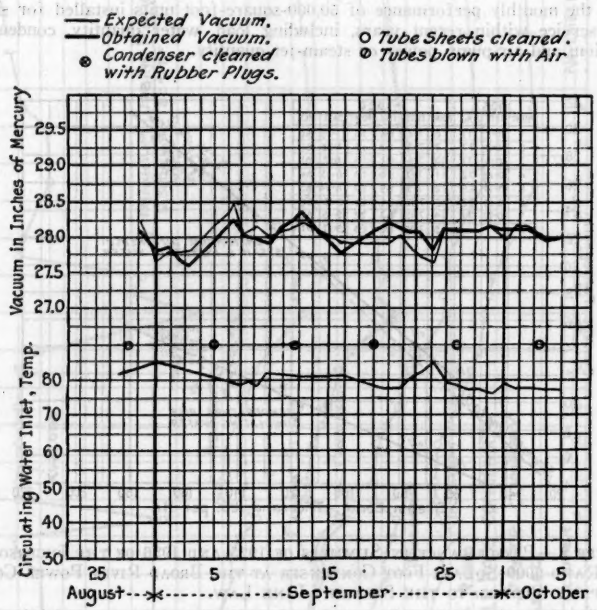


FIGURE 3.—APPROXIMATELY 40 DAYS' PERFORMANCE OF THE INGERSOLL-RAND 25,000-SQUARE-FOOT CONDENSER SERVING THE 38,000-KILOWATT (MAXIMUM RATED) TURBINE AT THE BROAD RIVER POWER COMPANY, PARR, S. C.

(This chart also was plotted by the operating department of the W. S. Barstow Management Association, Inc., similar to Figure 1.)

These operating readings were taken once each day. The daily performance is consistent with the test performance, but what is of more significance, the results after a month and a half were practically the same as on the day of the test.

Figure 6 shows operating and test readings taken from two months to two and one-half years after installation of a 14,100-square-foot unit at the City of Pasadena municipal plant. A significant feature of these results is the fact that excellent performance and high heat transfer are obtained at low velocities (see Table 1). The test results were obtained after "shooting" the tubes with corks to remove the slime.

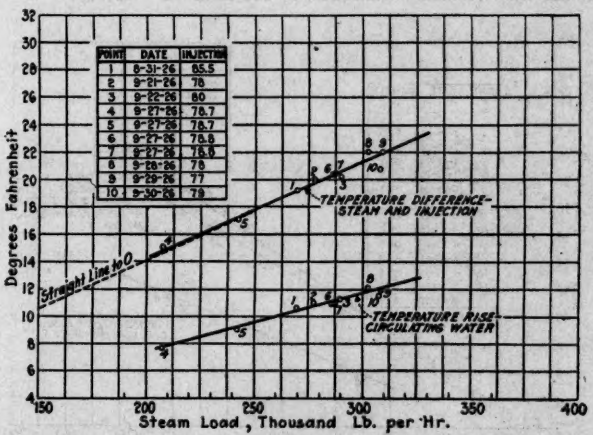

POINT | DATE | INJECTION
1 | 8-31-26 | 65.5
2 | 9-21-26 | 78
3 | 9-22-26 | 80
4 | 9-27-26 | 78.7
5 | 9-27-26 | 78.7
6 | 9-27-26 | 78.8
7 | 9-27-26 | 78.8
8 | 9-28-26 | 78
9 | 9-29-26 | 77
10 | 9-30-26 | 79
TEMPERATURE DIFFERENCE-STEAM AND INJECTION
TEMPERATURE RISE-CIRCULATING WATER
Straight Line 10
Degrees Fahrenheit
Steam Load, Thousand Lb. per Hr.

FIGURE 4.—STRAIGHT-LINE CHART SHOWING PERFORMANCE WITH WARM WATER OF THE INGERSOLL-RAND 25,000-SQUARE-FOOT CONDENSER SERVING THE 38,000-KILOWATT (MAXIMUM RATED) TURBINE AT THE BROAD RIVER POWER COMPANY, PARR, S. C.

(The chart shows the difference in temperature between the exhaust steam and the injection water; also the rise in temperature of the circulating water. The small table shows the actual injection temperature.)

CONDENSATE TEMPERATURE.

Figure 2 shows condensate temperatures at practically steam temperatures. The condenser shows a condensate temperature about 3 degrees under steam temperature. The two remaining condensers show very hot condensate, the 30,000-square-foot unit being fitted with a reheating hotwell. Detailed figures need not be given since, with modern stations and extraction heating, condensate temperature has little or no effect on plant economy. This is now widely recognized.*

* Refer to January, 1926, N.E.L.A. Report, page 1, paragraph 4. "With installations using stage bleeding, a moderate depression of hotwell temperature is of little consequence in the general heat balance, despite the frequent use of such differences as a basis of comparison of condenser results." It is important, however, that the condensate be low in oxygen, and this is obtained. In recent tests the condensate shows 0.05 cc. per liter.

CONDENSER EFFICENCY.

It has been frequently argued that no marked difference in efficiency could exist between different condensers, and that the only merit in the Ingersoll-Rand design was the use of a single pass, smaller tubes, higher velocity, excessive water, and large power. Velocities, frictions, and water quantities are given in Table 1 and are checked by temperature rise in the various charts. It will be noted that the velocities are little higher than the range of 5 to 8 feet per second employed in conventional condensers. The gain to be expected from velocity and tube size can readily be isolated, thus indicating what remaining gain must be the result of efficiency.

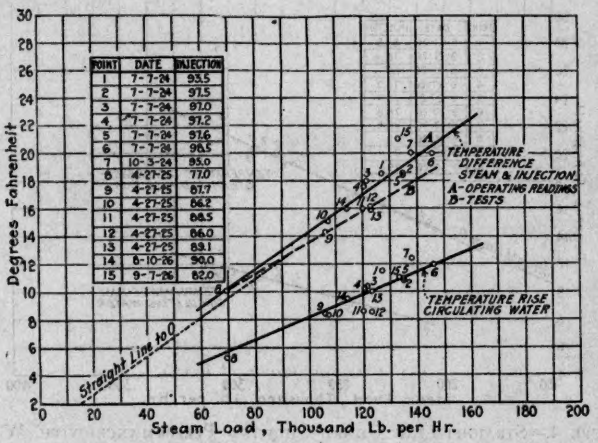


FIGURE 6.—STRAIGHT-LINE PLOTTING OF PERFORMANCE AND TEST OF INGERSOLL-RAND 14,100-SQUARE-FOOT CONDENSER, PASADENA, CAL.

(Condenser operates in connection with cooling tower and very warm water as shown by the table.)

From Figure 7, a two-pass condenser with 1-inch tubes and a single-pass condenser with ¾-inch tubes have the same friction at velocities of 6 and 8 feet per second, respectively. With these ratios of velocities and the same total water quantity, a brief calculation of number of tubes, areas for water flow, etc., shows that the single-pass unit with ¾-inch tubes would have about half the surface of the two-pass unit with 1-inch tubes. This is not strictly correct, but is sufficiently close to serve as an example without introducing unwieldly figures.

Due to the higher velocity, the ¾-inch tubes would have an advantage corresponding to $\sqrt{8/6}$ or 115 per cent in heat transfer. The further advantage of the ¾-inch diameter as against the 1-inch diameter would increase this to about 125 per cent.

If the single-pass condenser of half the surface is to do the same work, there must be about double the heat transfer per square foot. It follows,

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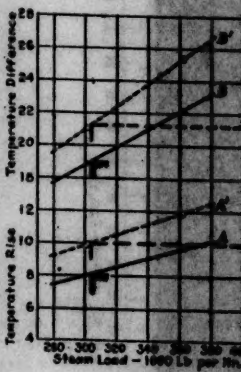

FIGURE 8.—T
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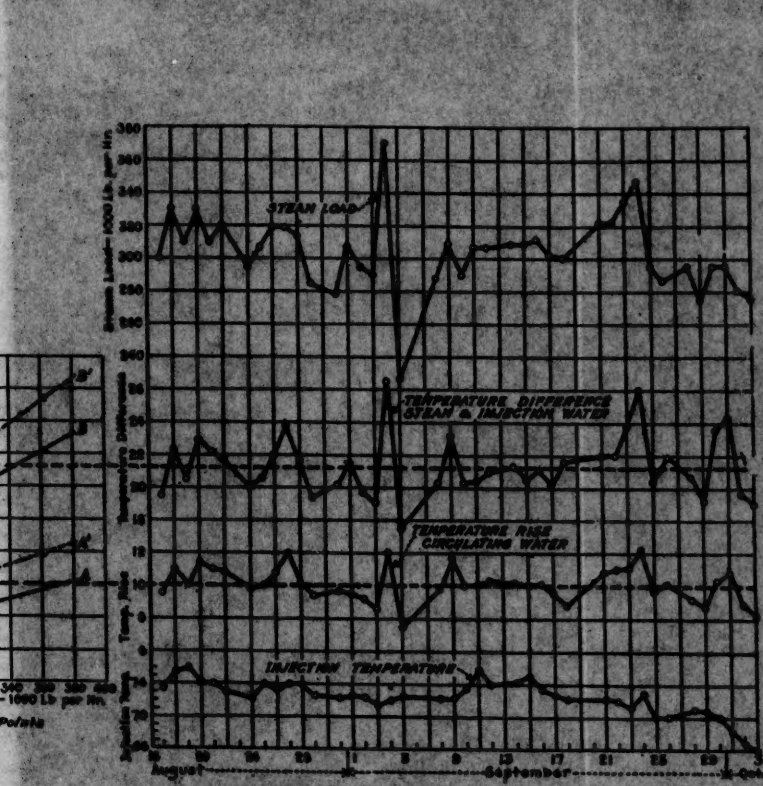
50-day period.



IGURE 8.—TEST PERFORMANCE AND 50 DAYS' OPERATING PERFORMANCE OF AN INGERSOLL-RAND 30,000-SQUARE-FOOT CONDENSER AT THE NEW YORK EDISON COMPANY, WATERSIDE STATION, SERVING A 35,000-KILOWATT NON-BLEEDING TURBINE.

(The test points showing temperature difference against load are shown y B at the left, with rated water 70,000 gallons corresponding to tempera- ure rise as shown by A. The condenser was operated with reduced water s shown by the graph of temperature rises. The test results, corrected for he reduced water, are shown by lines A′ and B′. The dotted lines at the ight of the chart show temperature rise and temperature difference ex- ected from tests, and are to be compared with actual results obtained for 0-day period.)

therefore, that as only 25 per cent gain can be expected from velocity and tube size, a large improvement must be obtained from internal efficiency. The efficiency must be increased to 200/125 or 160 per cent of what it was in the two-pass condenser. The principal gain must come from this source.

Condenser efficiency is the ratio of the overall heat transfer to that for some standard conditions of tube velocity, temperature, diameter, etc., or—

$$\text{Efficiency ratio} = \frac{\text{Overall heat transfer}}{\text{Single-tube heat transfer}}$$

The denominator is the heat transfer in a calorimeter, with a single tube taken from the condenser, tested at the same conditions of temperature,

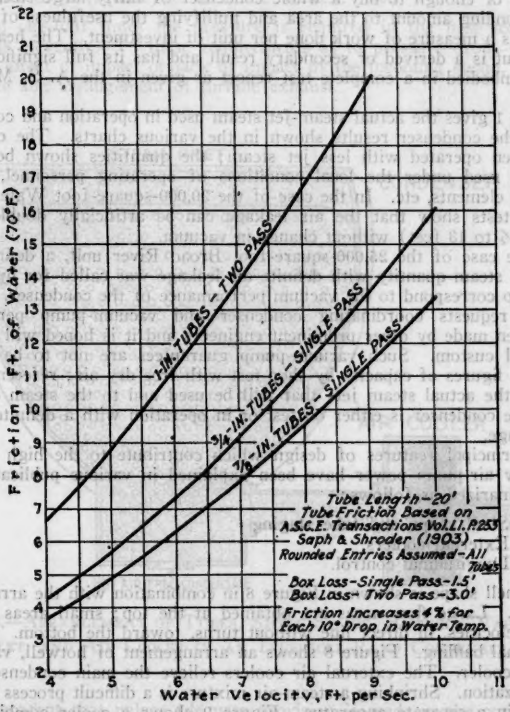

Friction, Ft. of Water (70°F.)
Water Velocity, Ft. per Sec.
1-IN. TUBES - TWO PASS
¾-IN. TUBES - SINGLE PASS
⅞-IN. TUBES - SINGLE PASS
Tube Length-20'
Tube Friction Based on
A.S.C.E. Transactions Vol. LI. P.253
- Saph & Shroder (1903)
Rounded Entries Assumed-All Tubes
Box Loss-Single Pass-1.5'
Box Loss- Two Pass-3.0'
Friction Increases 4% for
Each 10° Drop in Water Temp.

FIGURE 7.—COMPARISON OF THE FRICTION IN A 1-INCH TWO-PASS CONDENSER WITH THAT IN ¾-INCH AND ⅞-INCH SINGLE-PASS CONDENSERS.

(A 6-foot velocity in a 1-inch two-pass condenser gives the same friction as an 8-foot velocity in a ¾-inch single-pass condenser.)

velocities, cleanliness, etc. The ratio is analogous to the Rankine-cycle efficiency ratio of a turbine. It is a measure of what is accomplished with thousands of tubes as against one tube.

The efficiency ratio is a variable not only with design but with operating conditions. At zero load the vacuum cannot attain the theoretical value corresponding to water temperature on account of the air leakage. Hence the efficiency is zero and all efficiency curves rise from this point, attaining values depending on design.

Efficiency must be obtained with small air-pump power. Extra steam-jet quantities of 1000 to 2000 pounds are frequently used in test or even in operation, and to quote heat transfers as obtained under these conditions without mention of the steam quantity, as is the almost universal custom, is worthless. Such steam quantities when capitalized represent $15,000 to $30,000, or enough to buy a whole condenser of fairly large size, adding a corresponding amount to the area and nullifying the usefulness of the coefficient as a measure of work done per unit of investment. The heat-transfer coefficient is a derived or secondary result and has its full significance only when embodied in a complete test report as given in the A. S. M. E. Test Code.

Table 1 gives the actual steam-jet steam used in operation and corresponding to the condenser results shown in the various charts. The condensers have been operated with less jet steam; the quantities shown being those actually used under the local conditions of operating personnel, arrangement of elements, etc. In the case of the 30,000-square-foot Waterside condenser, tests show that the air leakage can be artificially doubled (from about 6½ to 13 feet) without change in vacuum.

In the case of the 25,000-square-foot Broad River unit, a definite guarantee of steam quantity with definite air leakage was called for by the purchaser to correspond to the vacuum performance of the condenser. Recently similar requests coördinating condenser and vacuum-pump performance have been made by other prominent engineers and it is hoped will become a universal custom. Such vacuum-pump guarantees are not to be confused with the figures of capacity by shop test with free dry air; reference is had here to the actual steam jets that will be used and to the steam consumed when the condenser is either on test or in operation with a definite limit of air leakage.

The principal features of design which contribute to the high efficiency with low air-pump power have been explained in various publications and are summarized as follows:

Shell shape and tube staging
External coolers
Longitudinal control.

The shell shape is shown in Figure 8 in combination with the arrangement of tubes. Large flow areas are obtained at the top; small areas and sustained velocities, in direct line without turns, toward the bottom. There is no internal baffling. Figure 8 shows an arrangement of hotwell, vapor outlet, and cooler. The external air coolers relieve the main condenser of air devaporization. Shrinking a steam-air mixture is a difficult process best performed in a separate apparatus. Figure 9 shows a cooler combined with steam jets. Figures 10 and 11 show how longitudinal control is obtained.

The interest shown in longitudinal distribution warrants a brief explanation. The condenser in large sizes is divided into four compartments by tight support sheets. Condensate is trapped to prevent flashing, and vapor and air are removed at different vacuums corresponding to the pressure-

drop requirements in each section. Each pair of vapor outlets is connected to one of two coolers and the primary steam jets on these operate at different vacuums. Between the two outlets of a pair, control is secured by an orifice plate located in the outlet which it is desired to choke.

Figure 12 shows the relative steam loads and pressure drops along the length of a condenser. As the inlet end has the colder water, it has greater condensing capacity than the discharge end. Comparing the first 2-foot section of a condenser with the eleventh or last 2-foot section of a 22-foot condenser, the relative capacities and pressure drops in a typical case are:

	First section	Eleventh section
Condensing capacity	1.88	1.00
Pressure drop (square)	3.50	1.00

Three and one-half times the pressure is required in the first section as against the eleventh. To these requirements as to pressure drops must be added the effect of impact or lack of impact of the incoming steam, depending on size and arrangement of turbine exhaust.

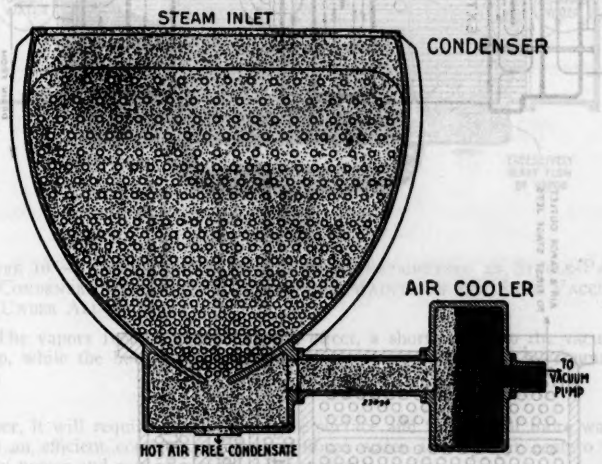


FIGURE 8.—ARRANGEMENT OF TUBES, SHELL SHAPE, HOTWELL, VAPOR OUTLET, AND AIR COOLER IN AN INGERSOLL-RAND CONDENSER.

It is apparent that unless adequate provision is made in the construction of the condenser and the air-removal system, the steam cannot be expected to penetrate completely except at the hot end of the condenser. In the remaining parts there will be varying degrees of penetration with corresponding waste of surface throughout the bottom. These conditions prevail even when a wedge-shaped shell is used and the air cooling is done externally.

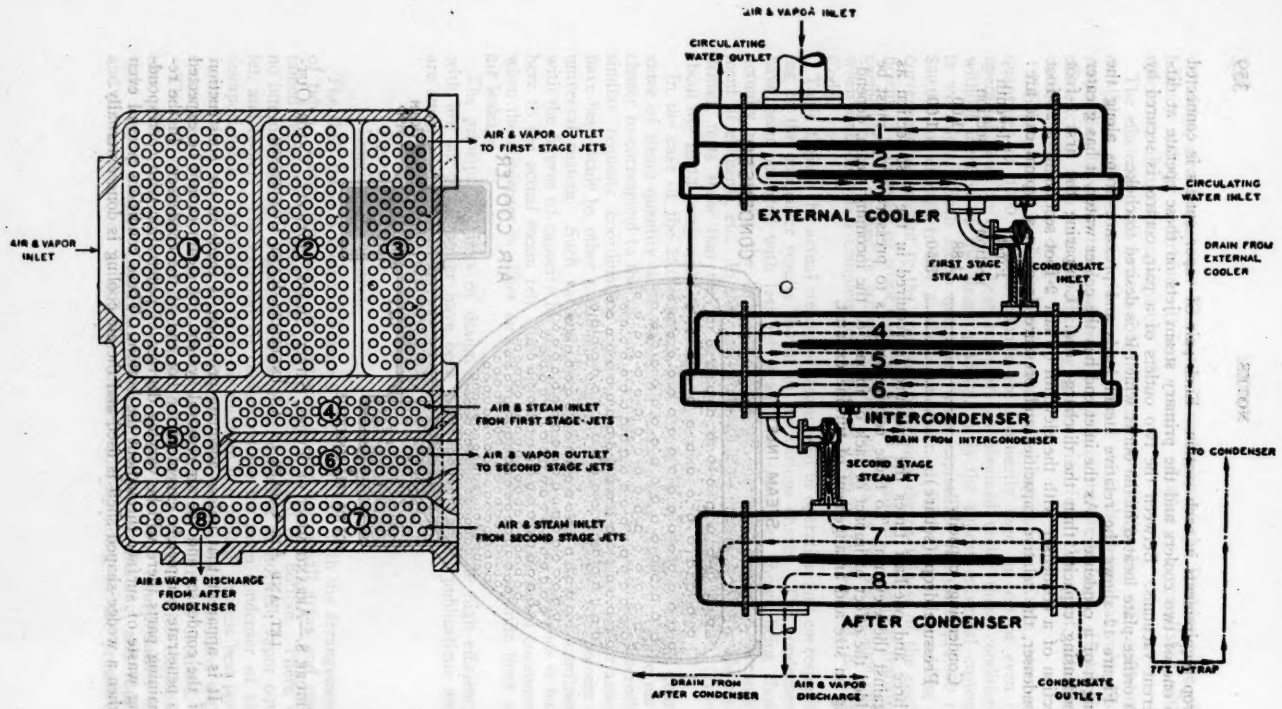


FIGURE 9.—ARRANGEMENT OF AIR DEVAPORIZER OR COOLER, TOGETHER WITH STEAM JETS, INTERCONDENSER, AND AFTERCONDENSER IN INGERSOLL-RAND COOLER DESIGN.

SINGLE-PASSING.

If a condenser is not efficient, with the result that there are large numbers of tubes which are wasted and with corresponding waste of water, the condenser had better be of the two-pass type, giving the water a second chance to be heated. It is axiomatic that the less efficient a condenser or water heater, the more times the water must be passed back and forth in order for it to do its work.

Assume a condenser with only 20 per cent of waste tubes and water, this being far from an unfavorable assumption.* If this is a single-pass con-

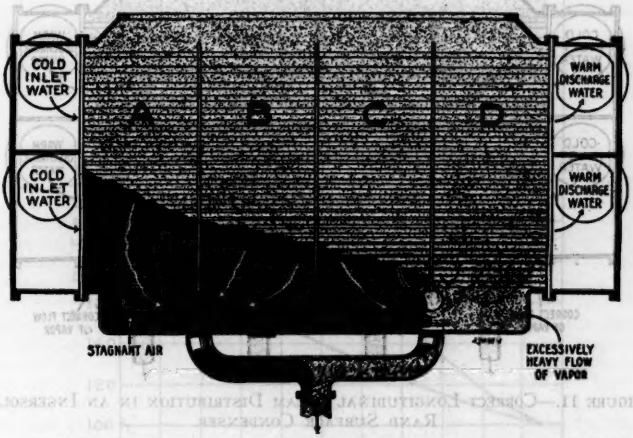


FIGURE 10.—FAULTY LONGITUDINAL STEAM DISTRIBUTION IN SINGLE-PASS CONDENSER WHEN THE VACCUM PUMP MAINTAINS THE SAME VACCUM UNDER ALL SECTIONS.

(The vapors from section D are, in effect, a short-circuit to the vacuum pump, while the bottoms of the other sections are surrounded by stagnant air.)

denser, it will require 25 per cent more surface and 25 per cent more water than an efficient condenser. The additional water adds 25 per cent to the pump power and cost of piping and tunnels.†

The water can be slowed down to the normal quantity, but this will reduce the condenser capacity so that more surface will be needed. This, however, means more tubes and will reduce the velocity still further, with corresponding reduction in coefficient. The result is shown in Figure 13. To obtain the same capacity 100 per cent more surface is needed.

* In the September, 1924, N.E.L.A. Report, p. 12, tests are reported showing very small relative rise in the first pass of a two-pass condenser and indicating between 1/5 and 1/10 the efficiency of heat transfer for the first half of the surface as against the second half. This corresponds to about 40 per cent waste surface.

† Various means of chocking this waste water such as valves, multiple-passing, or using smaller tubes were described some years ago and are covered in various patents.

This surprising result is due to the handicap of only 20 per cent waste tubes and water; there is only 80 per cent water to do all the work, calling for greater actual temperature rise and more and more surface with corresponding lower velocities, all of which finally comes to a balance as shown. Such a condenser had better be made of the two-pass type; and, conversely it is only when waste tubes are eliminated that condensers of the single-pass type are justified.

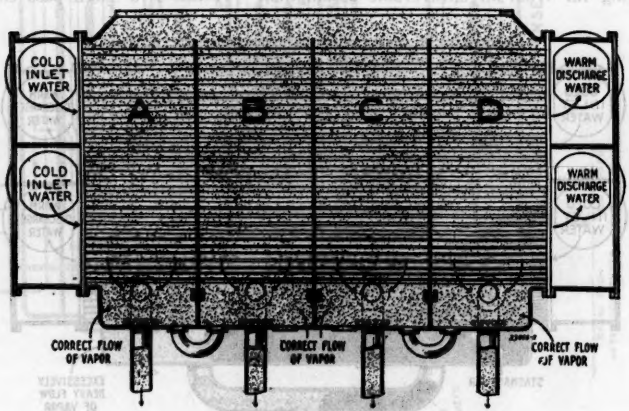


FIGURE 11.—CORRECT LONGITUDINAL STEAM DISTRIBUTION IN AN INGERSOLL-RAND SURFACE CONDENSER.

(Each section is made, in effect, a separate condenser by steam-tight support plates. Because each section has its separate vapor outlet, a different and suitable vacuum can be maintained under each section.)

APPENDIX NO. 1.

STRAIGHT-LINE LAW.

By referring to the fundamental equation of heat transfer from steam to water in a tube, it will be seen that the ratio of the temperatures

$$\frac{\text{Steam—cold water}}{\text{Steam—hot water}} = \text{Constant}$$

for any given condition of water flow and coefficient of heat transmission as determined by physical factors such as material of tube, speed of water, etc.* It follows by algebraic transformation that the ratio of the temperatures

$$\frac{\text{Hot water} - \text{cold water}}{\text{Steam—cold water}} = \text{Constant, also.}$$

* See Hirshfeld & Barnard's Elements of Heat-Power Engineering. p. 463.

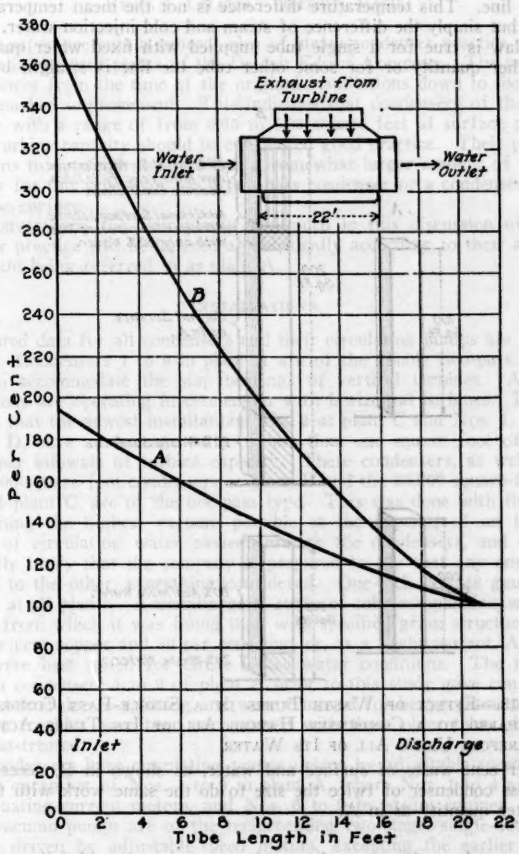


FIGURE 12.—LINE A SHOWS THE RELATIVE STEAM LOAD ALONG THE LENGTH OF A 22-FOOT CONDENSER IN ORDER TO GIVE EQUAL ACTIVITY THROUGHOUT THE DEPTH IN ALL LONGITUDINAL SECTIONS OF THE CONDENSER. THE FIRST 2-FOOT SECTION MUST CONDENSE 185 PER CENT AS MUCH STEAM AS THE LAST OR 11TH 2-FOOT SECTION. LINE B SHOWS THE NECESSARY PRESSURE DROPS TO INSURE FLOW OF THE STEAM QUANTITIES INDICATED BY A.

As the difference of the hot and cold water is the rise in temperature or load, it follows that the load plotted against the temperature difference is a straight line. This temperature difference is not the mean temperature difference but simply the difference of steam and cold injection water.

This law is true for a single tube supplied with fixed water quantity; at some other quantity or for some other tube the line is straight but at another slope. A condenser, made up of many tubes, should follow the same law. Actually it departs more or less from this law, the reason being that the performance of a single tube is controlled by thermal conditions whereas that of a condenser is influenced by flow and distribution forces as well. The flow conditions in turn are determined by steam volume, air leakage, capacity of the vacuum pump, and condenser design—particularly flow conditions through the tube banks and provisions for air devaporization. These have a complex influence on performance, which need not be discussed. The usefulness of the straight-line law lies in the fact that it is convenient for plotting, a straight edge being employed, and that it defines the regions of departure from the theoretical and best performances.

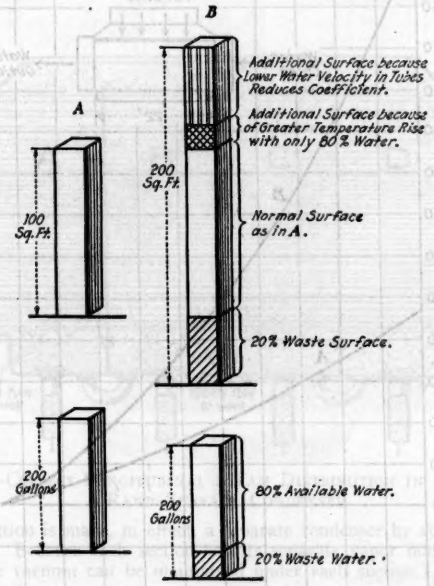


FIGURE 13.—EFFECT OF WASTE TUBES IN A SINGLE-PASS CONDENSER AS COMPARED TO A CONDENSER HAVING ALL OF ITS TUBES ACTIVE AND THEREFORE USING ALL OF ITS WATER.

(20 per cent waste of surface and water, as shown in B, necessitates a single-pass condenser of twice the size to do the same work with the same total water quantity as in A.

STEAM-CONDENSER PRACTICE AND PERFORMANCE.

By F. J. Chatel,[1] Detroit, Mich.

This paper presents a general idea of steam-condenser practice and performance in the four plants of the Detroit Edison Company. The condenser performances from the time of the original installations down to the present show a marked improvement. This indicates that condensers of the single-pass type with a range of from 0.95 to 1.05 square feet of surface per kilowatt of turbine capacity should be considered good practice. Their performance seems to outweigh the fact that a somewhat larger amount of water is necessary for this type than for a two-pass condenser or a condenser having more tube surface.

For convenience, the four plants mentioned in this discussion of steam-condenser practice will be lettered alphabetically according to their ages, the oldest plant being referred to as plant A.

INSTALLATIONS.

Tabulated data for all condensers and their circulating pumps are given in Table 1. Condensers 1 to 9 in plant A are of the double two-pass type designed to accommodate the step bearings of vertical turbines. All other installations are operating in connection with horizontal turbines. It should be noted that the newest installations, No. 2 at plant C and Nos. 1, 2, and 3 at plant D, have an average ratio of less than one square foot of cooling surface per kilowatt of turbine capacity. These condensers, as well as the two 70,000-square-foot condensers at plant B and the 35,000-square-foot condenser at plant C, are of the one-pass type. This was done with the object of obtaining the highest vacuum possible at the expense of an increased amount of circulating water passed through the condensers, and does not necessarily imply that the company is prepared to say that any one type is superior to the other, everything considered. One-inch No. 18 gauge tubes are used at all plants. A metallurgical study of tube composition was made in 1914, from which it was found that, with specified grain structures, tubes of 70 per cent copper and 30 per cent zinc, or, as a slight variant, Admiralty metal, were best suited for Great Lakes water conditions. The tubes installed in condensers 1 to 9 of plant A prior to this study gave considerable trouble at first due to splitting and were removed for heat treatment. Practically no troubles have been encountered with these older tubes since they were heat-treated.

All condensers have circulating pumps driven by adjustable-speed motors, with the exception of Nos. 1 to 9 in plant A, where Nos. 1 to 4 are driven by alternating-current motors, and Nos. 6 to 9 by steam engines.

Dry vacuum pumps are of the reciprocating two-stage, single-acting type, and are driven by adjustable-speed motors, excepting the earlier installations, Nos. 1 to 9 in plant A, where they are driven by steam engines.

Figures 1 to 9 show the different types of tube layouts at the four plants. The solid circles indicate plugged holes in the tube sheets, the tubes being omitted to provide steam lanes, and to accommodate drain plates and drain pipes. Figure 10 is a front elevation of condensers 1, 2, and 3, plant D, and is included to show the location of the gates.

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TABLE 1 MAIN CONDENSERS AND CIRCULATORS

Plant	Cond. No.	Put in service	Max. turbo-generator cap., kw.	Max. possible cooling surface, sq. ft.	Condenser: Cooling surface in service at time of tests, sq. ft.	Condenser: Sq. ft. surface per k.w. turbine cap.	Condenser: No. of passes	Condenser: No. tubes installed	Condenser: Length tubes bet. tube sheets	Condenser: Gage of tubes	Condenser: Tube sheet diam.	Tube composition: Cu.	Tube composition: Zn.	Tube composition: Sn.	Circulating pumps: Number	Circulating pumps: Size, in.	Circulating pumps: Estimated head, ft. (max.)	Circulating pumps: r.p.m. (max.)	Circulating pumps: Capacity, one pump, G.p.m.	Motors: Type	Motors: Hp.
A	1	1912	9000	16000	16000	1.78	2	3624	17' 3"	18	14' 6"	70	30	...	1	24	22	720	17000	d. c.	150
A	2	1912	9000	16000	16000	1.78	2	3624	17' 3"	18	14' 6"	70	30	...	1	24	22	720	17000	a. c.	150
A	3	1911	9000	16000	16000	1.78	2	3624	17' 3"	18	14' 6"	70	30	...	1	24	22	720	17000	a. c.	150
A	4	1911	9000	16000	16000	1.78	2	3624	17' 3"	18	14' 6"	70	30	...	1	24	22	720	17000	a. c.	150
A	6	1909	14000	25000	25000	1.79	2	6028	16'	18	14' 6"	70	30	...	1	30	20	250	25000	steam	210
A	7	1910	14000	25000	24832	1.74	2	5831	16'	18	14' 6"	70	30	...	1	30	20	250	25000	steam	210
A	8	1911	14000	25000	24475	1.75	2	4712	16'	18	14' 6"	70	30	...	1	30	20	250	25000	steam	210
A	9	1913	15000	22500	22456	1.48	2	5017	15'	18	14' 6"	70	40	...	1	30	26	250	30000	steam	300
A	10	1920	30000	42700	39617	1.32	2	8407	15'	18	15' 0"	70	30	...	2	36	23	440	30000	d. c.	300
B	1	1915	20000	35000	32500	1.625	2	6791	18' 3⅛"	18	14' 6"	70	30	...	1	42	16	360	36000	d. c.	250
B	2	1915	20000	35000	32457	1.624	2	6894	18' 3⅛"	18	14' 6"	70	30	...	1	42	16	360	36000	d. c.	250
B	3	1917	20000	35000	32140	1.622	2	6834	18' 3⅛"	18	14' 6"	70	30	...	1	42	16	360	36000	d. c.	250
B	4	1918	45000	70000	59210	1.316	2	9423	24' 3⅛"	18	16' 9"	70	30	...	2	48	22	300	60000	d. c.	450
B	5	1920	45000	70000	64600	1.435	2	10413	25' 3⅛"	18	16' 9"	70	30	...	2	48	22	300	60000	d. c.	450
B	6	1920	30000	42700	39660	1.320	2	8416	15'	18	15'	70	30	...	2	36	23	400	30000	d. c.	300
C	1	1922	10000	16300	16300	1.63	2	5294	11' 9⅛"	18	10' 10½"	70	30	...	1	30	26	400	20000	d. c.	182
C	2	1922	30000	35000	29101	0.97	1	5558	20'	18	Rect.	70	30	...	2	36	23	400	40000	d. c.	300
C	3	1923	10000	16300	16300	1.63	2	5294	11' 9⅛"	18	10' 10½"	70	30	...	1	30	26	400	20000	d. c.	182
D	1	1925	50000	52000	47000	0.95	1	7672	23' 3⅛"	18	17' 6"	70	29	1	2	48	20	255	60000	d. c.	255
D	2	1924	50000	52000	47300	0.95	1	7614	23' 3⅛"	18	17' 6"	70	29	1	2	48	20	255	60000	d. c.	255
D	3	1924	50000	52000	51800	1.04	1	8349	23' 3⅛"	18	17' 6"	70	29	1	2	48	20	255	60000	d. c.	255

Tube diameter, 1 in. in all cases.

Figure 1 shows the tube-sheet layout of No. 1 condenser in plant A. This is a double two-pass condenser and is similar in design to the others operating with the vertical turbines. The solid circles indicate the location of baffles in the lower half of the condenser. The tubes in the upper left-hand corner were removed to improve steam-entrance conditions. The open sections near the centers of the four panels are for staybolts in the water boxes. About 10 per cent of the tubes of these condensers are in the air coolers.

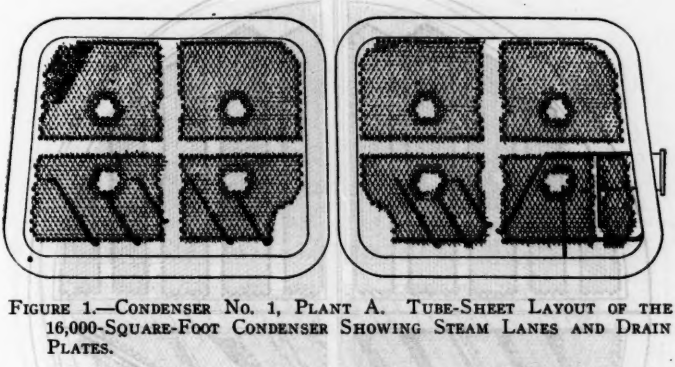

FIGURE 1.—CONDENSER NO. 1, PLANT A. TUBE-SHEET LAYOUT OF THE 16,000-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

Figure 2 shows the outline of the eccentric shell and the tube-sheet layout of No. 10 at plant A and No. 8 at plant B. In the upper half the original steam lanes were widened by omitting outer vertical rows of tubes on four of the tube banks. The lower half is divided into sections with drain plates and spouts extending to the bottom of the tube sheet in one quarter and to the air-cooler roof in the other quarter. Seventeen per cent of the total number of tubes are in the air coolers of these condensers.

Figure 3 shows the tube-sheet layout of condensers 1, 2, and 3 at plant B. As indicated by the solid circles, the steam lanes in the upper quarter over the air cooler are but one tube space in width, while the opposite quarter has tube lanes of varying widths. The air cooler contains 6.8 per cent of the tubes.

Figures 4 and 5 show the tube-sheet layouts of condensers 4 and 6, respectively, at plant B. These are one-pass condensers and have eccentric shells which provide a crescent-shaped steam belt on one side of the tube bundle. They are alike excepting as to the general arrangement of steam lanes and drain plates and the amount of cooling surface actually in service. No. 4 has 15.6 per cent of the tubes in the air cooler, while No. 6 has 14.8 per cent. Baffles are installed in the air coolers, so that the air, taking a zigzag course, will sweep all tubes at a high velocity.

Figure 6 shows the tube-sheet layout of the two 16,300-square-foot condensers operating at plant C in connection with 10,000-kilowatt turbines. The air coolers are equipped with baffles, and contain 22.6 per cent of the tubes.

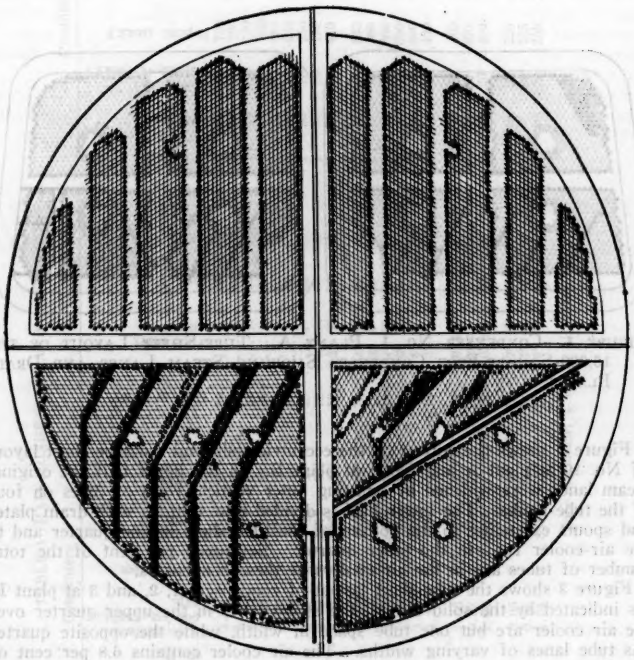

FIGURE 2.—CONDENSER NO. 10, PLANT A, AND CONDENSER NO. 8, PLANT B. TUBE-SHEET LAYOUT OF THE 42,700-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

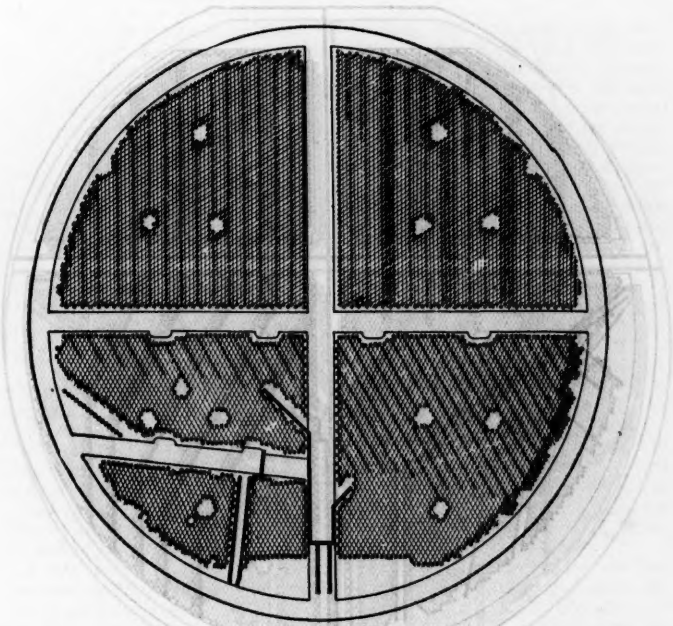

FIGURE 3.—CONDENSERS NOS. 1, 2, AND 3, PLANT B. TUBE-SHEET LAYOUT OF THE 35,000-SQUARE-FOOT CONDENSERS SHOWING STEAM LANES AND DRAIN PLATES.

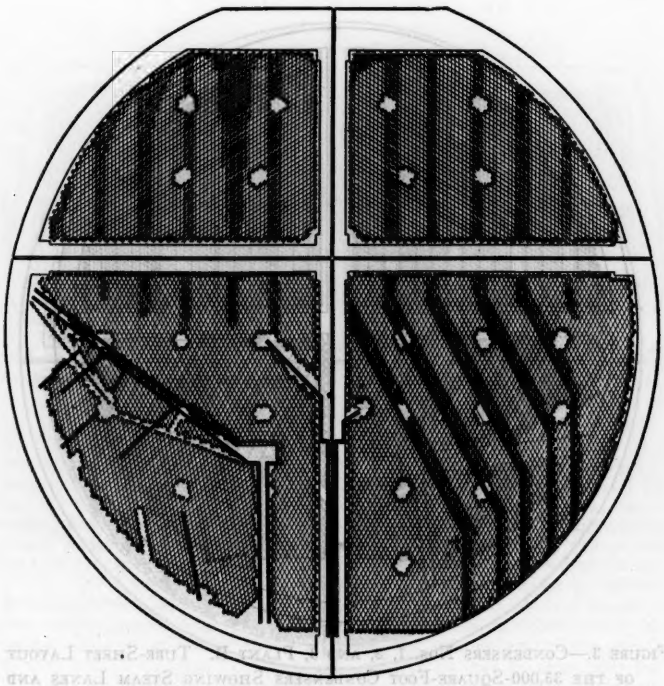

FIGURE 4.—CONDENSER NO. 4, PLANT B. TURE-SHEET LAYOUT OF THE 70,000-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

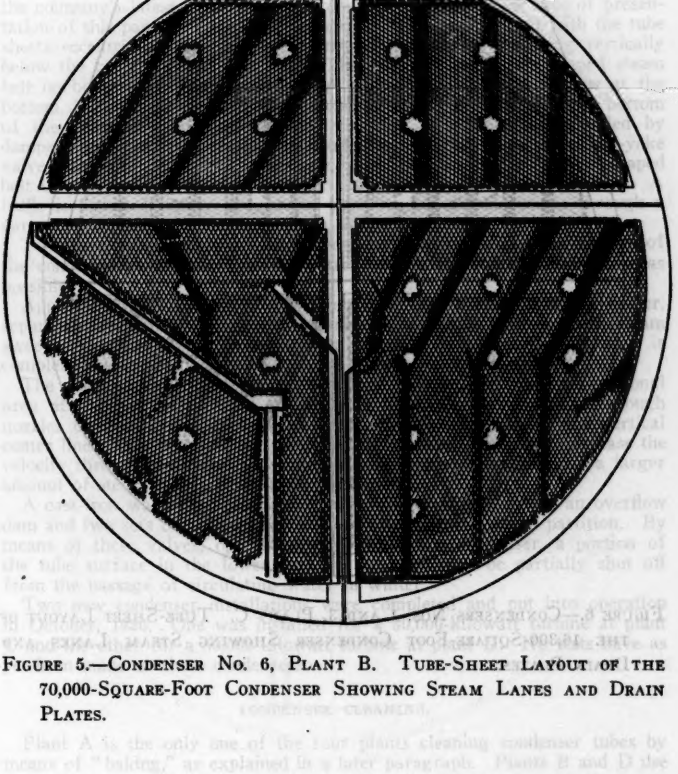

FIGURE 5.—CONDENSER NO. 6, PLANT B. TUBE-SHEET LAYOUT OF THE 70,000-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

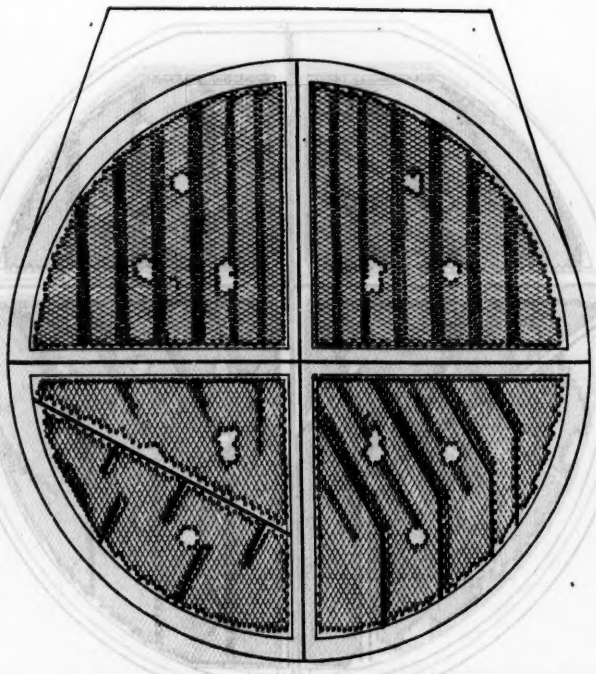

FIGURE 6.—CONDENSERS NOS. 1 AND 3, PLANT C. TUBE-SHEET LAYOUT OF THE 16,300-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

No. 2 condenser at plant C, shown in Figure 7, varies considerably from the other installations mentioned above in that it is rectangular in shape and the lanes between tube banks are wider at the top than at the bottom, giving a decreasing steam space. Tubes have been removed for steam lanes as shown by the solid circles. A "dummy," A, is located between the tubes in the central steam lane near the top of the tube bank. Below this is a series of troughs. The air outlet is near the top, there being 53.1 per cent of the tubes in the air cooler. The bowl-shaped hotwell is exceptionally large and acts as a deaerator.

Figure 8 is the tube-sheet layout of condenser No. 1, a single-pass condenser at plant D. This condenser and Nos. 2 and 3 at the same plant are the company's latest installations in regular operation at the date of presentation of this paper. The shell is of circular cross-section but with the tube sheets eccentric to the shell, the center of the tube sheets being vertically below the center line of the shell. This provides a crescent-shaped steam belt on both sides of the tube bundle, wide at the top and narrow at the bottom. On one side of this belt, steam is admitted directly to the bottom of the condenser, and reduces undercooling. The flow is controlled by dampers operated from outside the condenser by means of a screw-and-yoke valve mechanism. On the other side of the condenser the crescent-shaped belt is used as an equalizing chamber on the dry-vacuum-pump suction. A baffle is installed tightly against the tube bundle, and air passes through a narrow slot extending the full length of the condenser.

Tubes are arranged along radial lines, the lanes tapering from the top of the condenser to a point near the bottom where the spacing is as close as possible.

Approximately 40 per cent of the condenser is set apart as an air cooler, separated from the steam-condensing section by baffles. In summer, steam sweeps over into this air-cooling section, while in winter condensation is complete before the bottom of the condenser has been reached.

The cast-iron inlet water box is tapered, having greater cross-sectional area at the bottom. Water enters from both circulating pumps through nozzles directed vertically upward, both on the same side of the vertical center line. This directs more water to the top tubes, tending to increase the velocity through them and consequently insure the condensation of a larger amount of steam in this part of the condenser.

A cast-iron water box on the discharge side is provided with an overflow dam and two sets of butterfly valves located in the horizontal partition. By means of these valves, operable from outside the condenser, a portion of the tube surface in the lower part of the shell may be partially shut off from the passage of circulating water in winter.

Two new condenser installations were completed and put into operation in October, 1926. One was installed for a 30,000-kilowatt turbine at plant C and the other for a 50,000-kilowatt turbine at plant D. No tests have as yet been made on these condensers.

CONDENSER CLEANING.

Plant A is the only one of the four plants cleaning condenser tubes by means of "baking," as explained in a later paragraph. Plants B and D use cylindrical rubber plugs as scrubbers, while plant C uses for the same purpose disks of sheet-rubber packing ⅛ inch thick and slightly larger than the tube diameter. A nail is driven through the center of these disks to prevent their tipping over. These, like the cylindrical plugs, are forced through with general-service water at 100 pounds per square inch pressure.

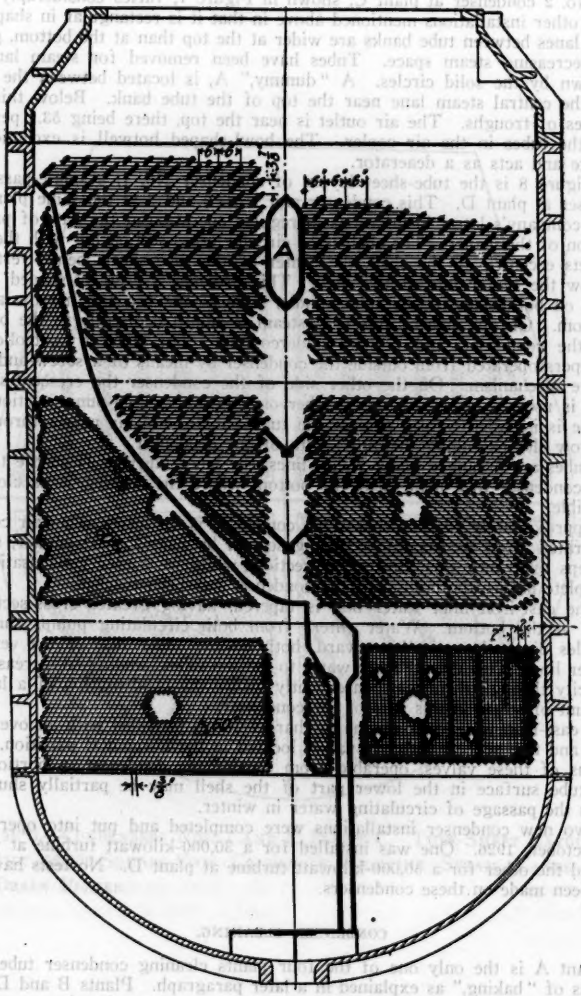

FIGURE 7.—CONDENSER NO. 2, PLANT C. TUBE-SHEET LAYOUT SHOWING STEAM LANES AND DRAIN PLATES.

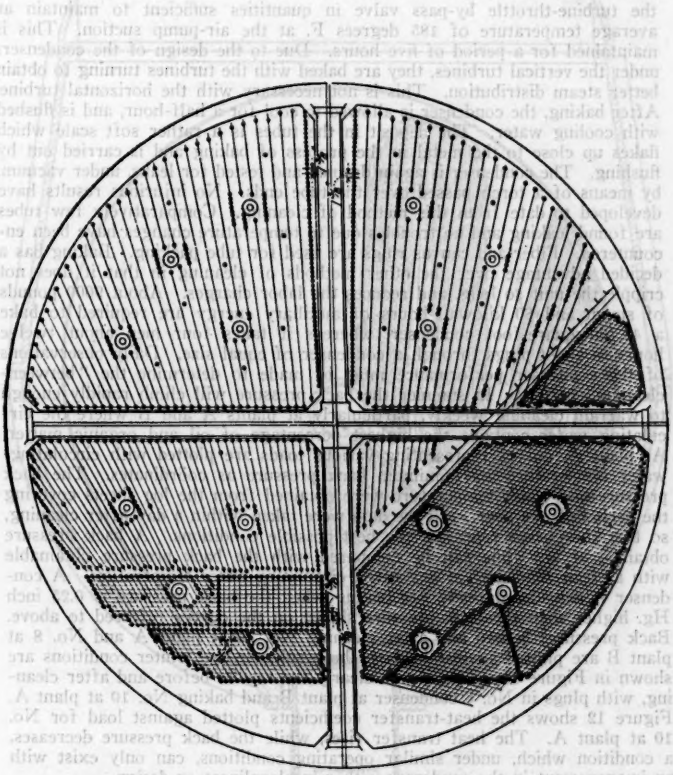

FIGURE 8.—CONDENSERS NOS. 1, 2, AND 3, PLANT D. TUBE-SHEET LAYOUT OF THE 52,000-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

In baking, such as is being done at plant A, the condenser is drained and the dry vacuum pump operated to maintain low vacuum. The priming valve between water box and steam space is cracked to allow any steam generated in the tubes to be drawn into the steam space. Steam is admitted through the turbine-throttle by-pass valve in quantities sufficient to maintain an average temperature of 185 degrees F. at the air-pump suction. This is maintained for a period of five hours. Due to the design of the condensers under the vertical turbines, they are baked with the turbines turning to obtain better steam distribution. This is not necessary with the horizontal turbine. After baking, the condenser is allowed to cool for a half-hour, and is flushed with cooling water. The deposit in the tubes is a rather soft scale which flakes up close to the metal in the process of baking and is carried out by flushing. The condenser is again drained and tested for leaks, under vacuum, by means of a torch passed over the tube ends. No injurious results have developed to date from this method of cleaning. Comparatively few tubes are found leaking and no troubles due to temperature changes have been encountered. Fiber and canvas rings are used for tube packing. Baking has a decided advantage over the other methods of cleaning in that it does not cripple the unit so long, and reduces the labor charges. About 9000 pounds of steam and 80 kilowatt-hours of auxiliary energy are required to bake a 42,700-square-foot condenser, whereas it takes four men about twelve hours to shoot plugs through a condenser of equal size. Daily observations of the condenser performance must be made to determine the "between-cleaning" periods. In summer the back pressure will climb rapidly enough to warrant cleaning weekly, particularly at plants A and B where the circulating water contains the highest percentage of oil and organic matter. At plant A, curves representing various loads are plotted with circulating-water temperature and minimum back pressure as coördinates. The back pressure and water temperature were obtained from the log sheets covering the period of one year. All the data were taken within a day after cleaning, so that the curves represent the best possible conditions. A back pressure obtained at any time can be compared with the back pressure obtainable with a clean condenser at any given circulating-water temperature. A condenser is considered ready for baking when the back pressure is 0.25 inch Hg. higher than the back pressure shown on the curves referred to above. Back pressures before and after cleaning No. 10 at plant A and No. 8 at plant B are plotted against load for both summer and winter conditions are shown in Figure 11. This shows clearly the results before and after cleaning, with plugs in No. 8 condenser at plant B and baking No. 10 at plant A. Figure 12 shows the heat-transfer coefficients plotted against load for No. 10 at plant A. The heat transfer rises, while the back pressure decreases, a condition which, under similar operating conditions, can only exist with an improvement in the condenser, either in cleanliness or design.

Comparing these sets of curves, it is evident that the drop in back pressure due to cleaning is very pronounced, but the back pressures of No. 8, plant B, either before or after cleaning are relatively higher than those of No. 10, plant A. This indicates that the tubes do not reach the same degree of cleanliness, as the condensers are of the same size. These results, however, do not take circulating-water temperature rise and pounds of condensate into account, but No. 10 turbine at plant A receives initial steam at a lower temperature and pressure and consequently has a higher water rate. Both have adjustable-speed motor-driven circulating pumps, making it possible to regulate their speeds to obtain the proper circulating-water temperature rise.

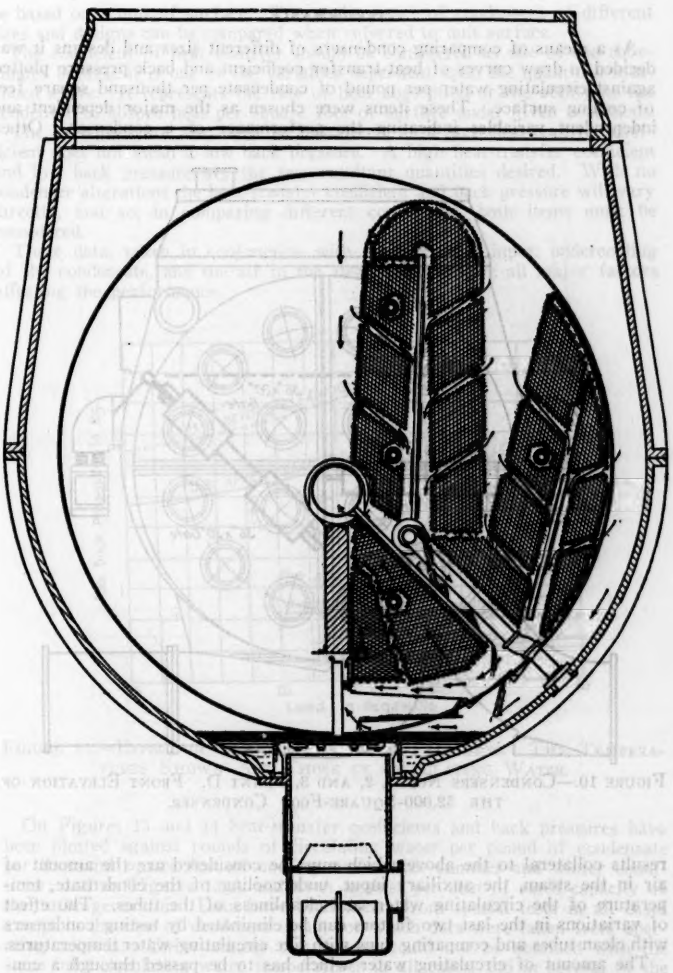

FIGURE 9.—CONDENSER NO. 4, PLANT D. TUBE-SHEET LAYOUT OF THE 52,000-SQUARE-FOOT CONDENSER SHOWING STEAM LANES AND DRAIN PLATES.

PERFORMANCE.

As a means of comparing condensers of different sizes and designs it was decided to draw curves of heat-transfer coefficient and back pressure plotted against circulating water per pound of condensate per thousand square feet of cooling surface. These items were chosen as the major dependent and independent variables indicating the performance of a condenser. Other

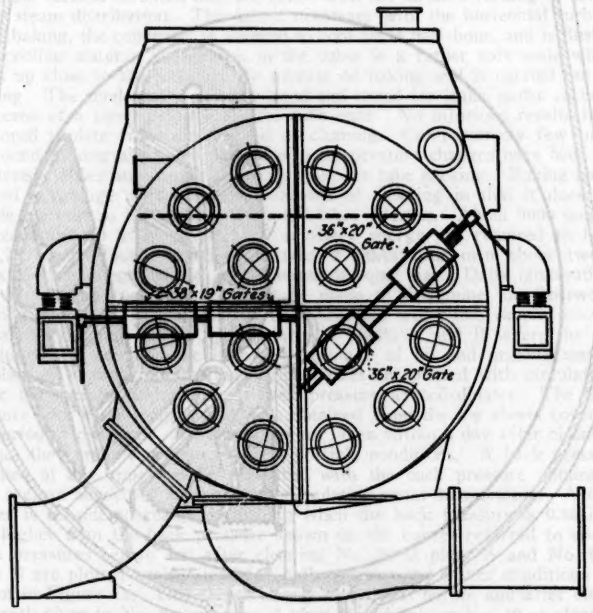

FIGURE 10.—CONDENSERS NOS. 1, 2, AND 3, PLANT D. FRONT ELEVATION OF THE 52,000-SQUARE-FOOT CONDENSER.

results collateral to the above which must be considered are the amount of air in the steam, the auxiliary input, undercooling of the condensate, temperature of the circulating water, and cleanliness of the tubes. The effect of variations in the last two factors can be eliminated by testing condensers with clean tubes and comparing runs with like circulating-water temperatures.

The amount of circulating water which has to be passed through a condenser to condense a pound of steam depends largely upon the design of the condenser, and is an indication of the ability of the unit as a whole to do its work, that is, to condense steam and create a vacuum. This indication, however, is not decisive, for a condenser may have an excessive amount of cooling surface and would appear very efficient if the surface were not taken into account. The circulating water per pound of condensate, then, should

be based on a unit of surface. The performance of condensers of different sizes and designs can be compared when referred to unit surface.

The coefficient of heat transfer should be considered as one of the coördinates because it is one of the variables dependent upon the quantities mentioned above. It represents the rate of heat absorption or the ability of the condenser to absorb heat per square foot of surface under given conditions.

The back pressure also must be considered, as a high heat-transfer coefficient does not mean a low back pressure. A high heat-transfer coefficient and low back pressure are the two resultant quantities desired. With no condenser alterations the heat-transfer coefficient and back pressure will vary directly, and so, in comparing different condensers, both items must be considered.

These data, taken in conjunction with the circulator input, undercooling of the condensate, and the air in the steam, will involve all major factors affecting the performance.

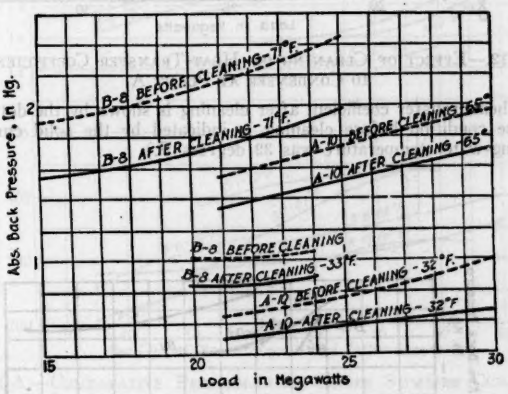

FIGURE 11.—EFFECT OF CLEANING ON BACK PRESSURES. THE TEMPERATURES SHOWN ARE THOSE OF CIRCULATING WATER.

On Figures 13 and 14 heat-transfer coefficients and back pressures have been plotted against pounds of circulating water per pound of condensate per thousand square feet of cooling surface for summer and winter conditions; and on Figures 13-A and 14-A against per cent of rated capacity of the turbo-generator. Data have been taken from special tests in all cases except in the case of condenser A-10, for which data were taken from operating records. Unfortunately test results could not be obtained with the same inlet circulating water temperatures in all cases, and in several of the tests air quantities removed were not measured. Air quantities measured in several condenser tests show an average range of from 4 to 8 cubic feet per minute. Figure 15 shows the percentage of main turbo-generator output used by the circulating pumps, and Figure 16 the undercooling of the condensate. Both are plotted against circulating water per pound of condensate per thousand square feet of cooling surface.

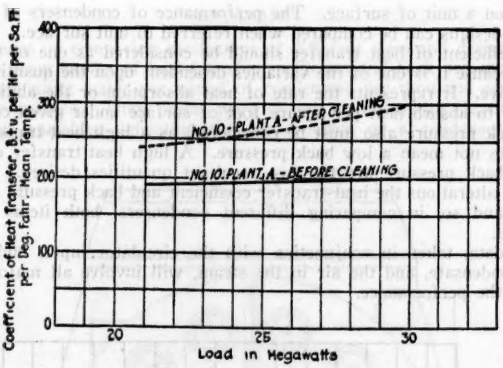

FIGURE 12.—EFFECT OF CLEANING ON HEAT-TRANSFER COEFFICIENT OF NO. 10 CONDENSER AT PLANT A.

(The heat-transfer coefficient after cleaning is shown by the dotted curve, while the condition before cleaning is indicated by the solid curve. The circulating-water temperature was 32 degrees F.)

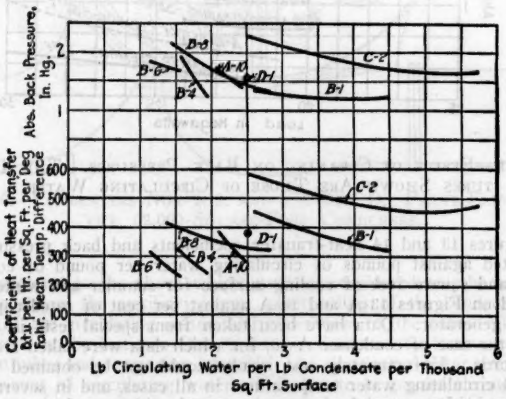

FIGURE 13.—COMPARATIVE PERFORMANCES UNDER SUMMER CONDITIONS.

(The letters and figures indicate the plant and condenser. The circulating-water temperatures are as follows: A-10 = 64 degrees F.; B-1 = 67 degrees F.; B-4 = 73 degrees F.; B-6 = 76 degrees F.; B-8 = 66 degrees F.; C-2 = 70 degrees F.; D-1 = 73 degrees F.)

DISCUSSION OF DATA

From the curves on Figures 13 to 16, inclusive, and the data in Tables 1 and 2 it is possible to obtain an idea as to the relative merits of condensers. As an example, it is readily seen that No. C-2, although having a high rate of heat transfer, also has a high back pressure and gives less satisfactory results than the remaining condensers. Its abscissas are large quantities as compared with the others, with the exception of B-1. This is due to the small cooling surface, 0.97 square feet per kilowatt of turbine capacity, resulting in a large ratio of circulating water per pound of condensate. The

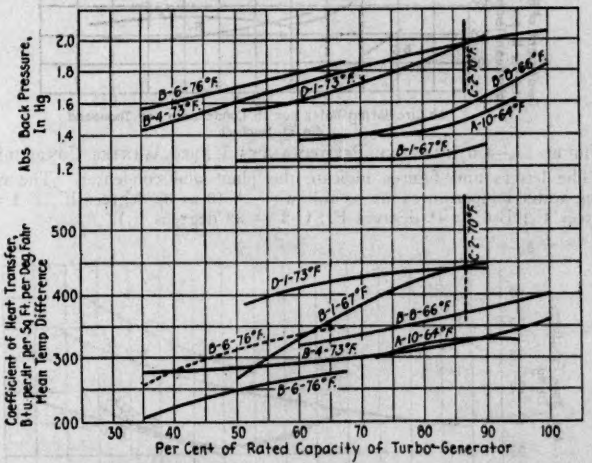

FIGURE 13-A.—COMPARATIVE PERFORMANCES UNDER SUMMER CONDITIONS.

(Back pressures and heat-transfer coefficients based on per cent of rated capacity of turbo-generators.

Dotted lines indicate heat-transfer coefficients for C-2 and B-6 with the mean temperature difference computed by the logarithmic formula. Solid lines for C-2 and B-6 indicate heat transfers computed by the arithmetical mean temperature difference.

The letters and figures indicate the plant, condenser and inlet circulating-water temperatures.)

circulating-pump power input is not abnormally high, but, on the other hand, undercooling of the condensate averages higher than the other installations.

Condenser B-1 although having large abscissas, operates with satisfactory heat-transfer coefficients and back pressures. Since it has a large amount of surface, 1.59 square feet per kilowatt of turbine capacity, it is evident that the quantity of circulating water per pound of condensate must be large. Its circulating-pump input is not abnormally large, but the undercooling is high during winter operation.

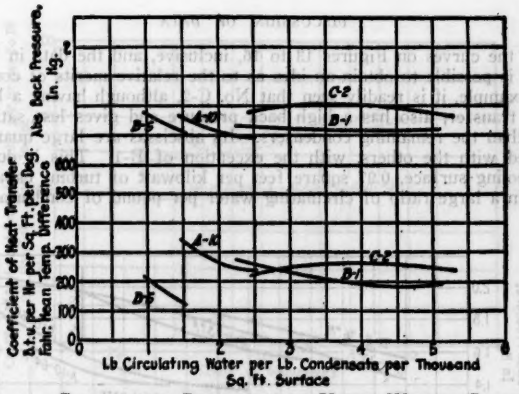

FIGURE 14.—COMPARATIVE PERFORMANCES UNDER WINTER CONDITIONS.

(The letters and figures indicate the plant and condenser. The circulating-water temperatures are as follows: A-10 = 32 degrees F.; B-1 = 33 degrees F.; B-6 = 41 degrees F.; C-2 = 32 degrees F.)

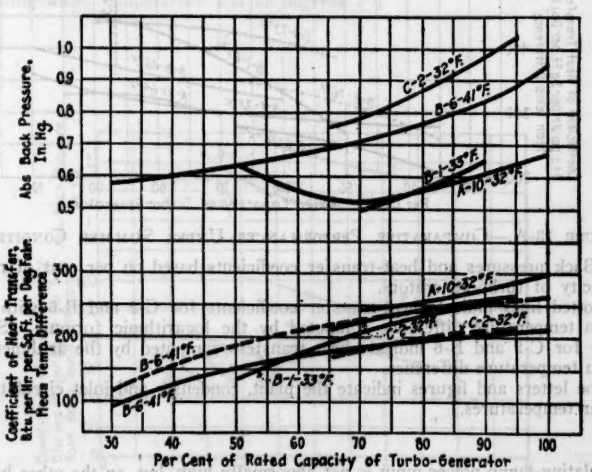

FIGURE 14-A.—COMPARATIVE PERFORMANCES UNDER WINTER CONDITIONS.

(Back pressures and heat-transfer coefficients based on per cent of rated capacity of turbo-generators.

Dotted lines indicate heat-transfer coefficients for C-2 and B-6 with the mean temperature difference computed by the logarithmic formula. Solid lines for C-2 and B-6 indicate heat transfers computed by the arithmetical mean temperature difference.

The letters and figures indicate the plant, condenser and inlet circulating-water temperatures.)

XU

TABLE I CORRELATION DATA AND RESULTS

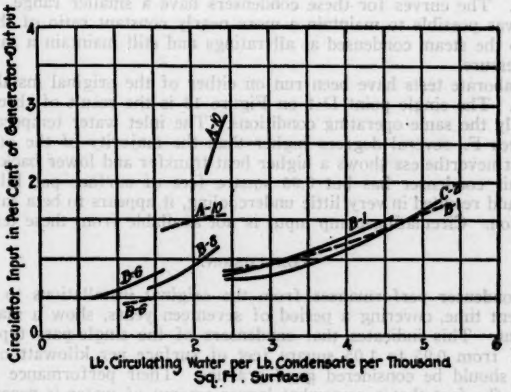

FIGURE 15.—POWER INPUT OF CIRCULATING-PUMP MOTORS.

(Solid lines indicate summer conditions—circulating water at 66-70 degrees F. Dotted lines indicate winter conditions—circulating water at 32-34 degrees F.)

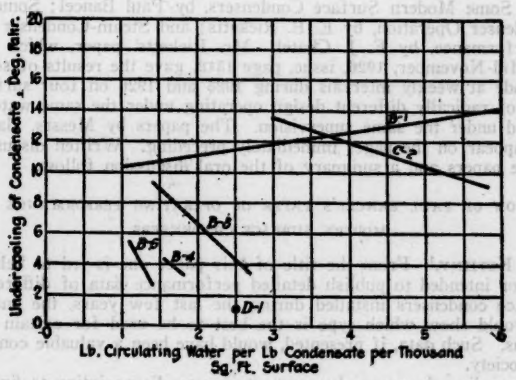

FIGURE 16.—UNDERCOOLING OF CONDENSATE.

The remaining curves, having smaller abscissas, indicate more surface per kilowatt of capacity or less cooling water per pound of condensate. See Table 2. The curves for these condensers have a smaller range, indicating that it was possible to maintain a more nearly constant ratio of the cooling water to the steam condensed at all ratings and still maintain a satisfactory back pressure.

No elaborate tests have been run on either of the original installations at plant D. The single point D-1 on Figure 13 is the result of three runs at practically the same operating conditions. The inlet water temperature was 73 degrees F., several degrees higher than the majority of the others outlined, but nevertheless shows a higher heat transfer and lower back pressure. Since this condenser has but 0.95 square feet of surface per kilowatt installed, and resulted in very little undercooling, it appears to be a satisfactory installation. Circulating-pump input is not available from these tests.

CONCLUSION.

The condenser performances from the original installations to those of the present time, covering a period of seventeen years, show a marked improvement. This indicates that condensers of the single-pass type with a range of from 0.95 to 1.05 square feet of surface per kilowatt of turbine capacity should be considered good practice. Their performance seems to outweigh the fact that a somewhat larger amount of water is necessary for this type of apparatus than for a two-pass condenser or a condenser having more tube surface.

DISCUSSION AT SESSION ON CENTRAL-STATION POWER.

Three papers were presented at this session: namely, Operating Performance of Some Modern Surface Condensers, by Paul Bancel; Some Results of Condenser Operation, by E. B. Ricketts; and Steam-Condenser Practice and Performance, by F. J. Chatel. Mr. Ricketts' paper, which appeared in the Mid-November, 1926, issue, page 1312, gave the results of some fifty tests made at weekly intervals during 1925 and 1926 on four surface condensers of radically different design operating under the same water conditions and under the same supervision. The papers by Messrs. Bancel and Chatel appear on the pages immediately preceding. Written discussions on the three papers and a summary of the oral discussion follow.

DISCUSSION OF PAUL BANCEL'S PAPER OF OPERATING PERFORMANCE OF SOME MODERN SURFACE CONDENSERS.

G. L. KOTHNY.[1] From the title of this paper one is led to believe that the author intended to publish detailed performance data of different types of surface condensers installed during the last few years, the analysis of which would show which type is the best to be used for certain existing conditions. Such data, if presented, would have been a valuable contribution to the Society.

When reading the paper, however, it is very disappointing to find that it deals only with one type of condenser, built by the firm with which the author is connected.

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[1] Executive Engineer, C. H. Wheeler Mfg. Co., Philadelphia, Pa. Mem. A.S.M.E.

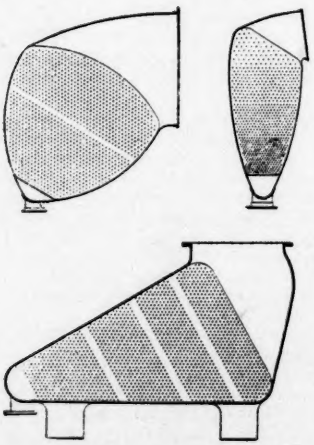

FIGURE 1.—TUBE-SHEET LAYOUTS FROM LORD WEIR'S PAPER.

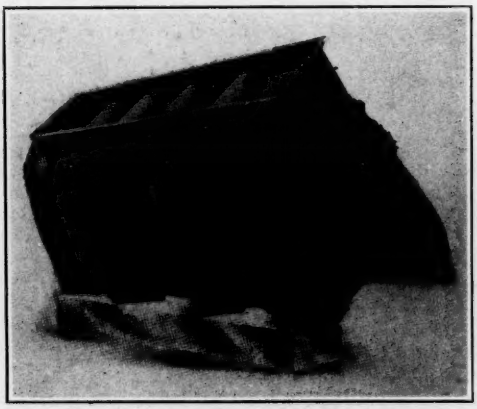

FIGURE 2.—WEIR-UNIFLUX SINGLE-FLOW CONDENSER.

The synopsis of the paper brings out four cardinal points, the first one being that a modern condenser requires about half the cooling surface of one of the conventional type. Let us analyze this. If it is true, then single-pass surface condensers used as long as 15 years ago by the United States and the British Navy on destroyers and cruisers must have been far advanced in their times. I would like to refer to a paper by William Weir, now Lord Weir, read in 1912 before the Institution of Engineers and Shipbuilders in Scotland. In this paper most of the principles which Mr. Bancel presents to the Society are already mentioned. Diagrams of tube-sheet layouts similar to the Ingersoll-Rand condenser are shown, as may be seen in Figure 1. Figure 2 shows a photograph of a Weir-Uniflux single-flow condenser in which the longitudinal control of the steam flow can be clearly seen.

Table 1 of Mr. Weir's paper, giving typical uniflux-condenser data and performance, installation N.o. 7 of the year 1912, records the following: Horsepower, 17,000 (12,700 kilowatts); steam consumption, 240,000 pounds per hour; cooling surface, 9000 square feet of ⅝-inch O. D. tubes 15 feet long; condensation rate, 26.7 pounds per square foot; vacuum obtained, 28.55 inches with 45 degrees F. inlet and 78 degrees F. discharge water temperature; hotwell temperature, 78 degrees F.; type of air pump, dual-twin beam. The temperature rise being 33 degrees F., the circulating-water quantity is calculated at 13,800 gallons per minute. The number of ⅝-inch tubes is 3670, giving a water velocity through the tubes of about 5.4 feet per second and a friction head of 6 feet.

These results compare very favorably with those published in Figure 2 of Mr. Bancel's paper. For the same discharge water temperature, viz., 78 degrees F., the same cooling surface, 9000 square feet, for a load of 15,200 kilowatts or 158,000 pounds of steam per hour, with an inlet temperature of 64 degrees F. and a circulating-water quantity of 22,000 gallons per minute, a vacuum of only 28.55 inches is obtained with the modern-type condenser at the Parr River 1925 installation.

The mean temperature difference in the Weir installation is 26 degrees F. and that in the Parr River installation is 19 degrees F. The heat transfer is 880 B.t.u. in the Ingersoll-Rand and 975 B.t.u. in the Weir condenser. The friction head in the Weir condenser is 6 feet and that in the Ingersoll condenser is 14.5 feet, and the latter uses 1.6 times the amount of cooling water that the Weir condenser does.

It would seem that the Weir condenser of 1912, in spite of its handicap in not having an efficient air-removal apparatus nor multiple air pumps working at different pressures, nor having condensate traps to prevent flashing nor orifices for choking, gave better results at less operating expense than the modern 1925 Ingersoll-Rand condenser.

Condensers similar to the Weir-Uniflux type, with longitudinal steam distribution, have been in general use in marine installations, especially for cruisers and destroyers. Mr. Bancel deserves credit for the initiative of applying marine practice to stationary installations. There is no doubt that in some of these the application of marine practice will be beneficial. To suggest (by implication) that the two-pass-type condenser be sent to the British Museum is not sound advice, as the selection of a type of condenser is based upon the capitalized cost of the performance, operation, and maintenance. Heat-transfer efficiency ratio, cooling surface, etc., are only incidentals to this major consideration.

Mr. Bancel claims that there is no essential difference between operating and test performance. There should not be if the condenser is cleaned constantly to keep it in test condition. Figure 1 in the paper indicates that the

Ingersoll-Rand condenser at Parr was cleaned weekly. In another paper, presented by a condenser operator at this meeting, it is shown that the modern single-pass condenser required three sandblastings and fifteen washings, while a two-pass condenser was sandblasted and washed only once during the same period.

Point three of the synopsis is fully answered by the data given of the Weir condenser performance aforementioned, the 1912 Weir single-pass condenser showing better overall performance results than the 1925 Ingersoll-Rand condenser. This is not surprising, since Mr. Bancel chokes the air outlets by orifice plates and operates the steam jets at the different vacuums. Unless the tubes are packed tight by stuffing boxes into the support plate, the theory of having different absolute pressures in the four compartments will not work well. He carefully omitted to mention how the tubes are made tight in the support sheets.

It is also difficult to understand why different pressures in sections A, B, C and D would not equalize on top of the condenser, even if the tubes are tight in the support sheets. After all, the total pressure in the condenser is the sum of the vapor tension and the partial air pressure. Since the temperature of the air-vapor mixture determines the vapor pressure and since this temperature is lower in section A, a more rapid flow of the mixtures would result, and there will be no stagnant air as shown in Figure 13. External air coolers are not novel and were extensively used more than twenty years ago by different condenser manufacturers.

The friction curves given in Figure 10 and based on Saph and Shoder's data (1903), while fairly accurate for the ⅞-inch and ¾-inch tubes, give higher friction values for the 1-inch tubes than are actually obtained or calculated, based on later researches by Stanton and Pannell (London, 1914), by W. L. DeBaufre and Milton C. Stuart (Trans. A. S. M. E., 1919), and by Walker, Lewis, and McAdams (1923). A 6-foot velocity in a 1-inch tube in a two-pass condenser would give the same friction as a 7.2-foot velocity in a ¾-inch tube in a single-pass condenser, and not 8 feet as shown by Mr. Bancel.

His statement that the ¾-inch tube would have an advantage corresponding to $\sqrt{8/6}$ over the 1-inch tube in heat transfer is incorrect, not only as far as the numerator is concerned, but also as far as its application is concerned. The change in water velocity affects the individual heat-transfer coefficient from metal to water film, but not the overall heat-transfer coefficient, of which Mr. Bancel speaks. This is explained in detail in another paper read by Messrs. McAdams, Sherwood, and Turner at this meeting.

P. E. Reynolds and S. W. Croll.[2] We have read Mr. Bancel's paper with a great deal of interest and if we allowed ourselves to follow his reasoning would be tempted to come to the conclusion that the single-pass condenser is the only efficient and desirable one. Furthermore it seems to us that the presentation of the excellence of the Ingersoll-Rand design takes precedence over the discussion of engineering features.

Mr. Bancel admits his original claim that a single-pass condenser having half the surface of a two-pass condenser could do the same work would be challenged, and we are inclined to think that this claim should still be challenged. Naturally if a single-pass condenser with half the surface does the same work as a two-pass condenser, the heat transfer of the former must be twice that of the latter and there are many data to show that this is not a fact. Numerous tests on two-pass condensers have shown heat transfers

[2] Respectively Vice-President and Mechanical Engineer, and President, Croll-Reynolds Co., Inc., New York. Mem. A.S.M.E.

as high as 700 B.t.u. and even higher, using comparatively cold water, and we find no tests on single-pass condensers which show heat transfers very much higher than these figures.

A test run this past summer on a 52,000-square-foot two-pass surface condenser at Waukegan, Ill., condensing 286,500 pounds of steam per hour and with inlet circulating water at 72 degrees F. showed the vacuum maintained was 28.85 inches of mercury. The rate of heat transfer on this particular condenser figures over 700 B.t.u., and incidentally the readings were taken using two mercury columns, two barometers, and checked by an absolute gauge. The condensate was measured by a venturi meter. The measured air leakage was approximately 7 cubic feet per minute. The results obtained with the single-pass Ingersoll-Rand condenser during August, 1925, as given in Mr. Ricketts' paper, show that this condenser with 72 degrees F. inlet water and with a rate of approximately 280,000 pounds of steam per hour maintained an average vacuum of 28.23 inches Hg, with a heat transfer rate of 496 B.t.u. Obviously these results do not show the superiority of the single-pass condenser claimed by Mr. Bancel.

Mr. Bancel brings out the fact, and we agree with him, that it is important to know how much steam is used by the air-removal apparatus. In the case of the 52,000-square-foot condenser mentioned above, the steam consumption of the steam-jet air-removal apparatus was 1100 pounds per hour. This, we think, compares very favorably with the 1800 pounds of steam per hour used with the 30,000-square-foot Ingersoll-Rand condenser at the New York Edison Company's Waterside Station.

We would also refer to tabulated operating data given in Mr. Ricketts' paper which give a comparison of operating results on a number of different types of condensers. We understand that the single-pass condenser mentioned in that paper marked "B" is one of the Ingersoll-Rand Company's condensers mentioned in Mr. Bancel's paper. A comparison of data obtained on this single-pass condenser with those on the two-pass condenser marked "D" would indicate that the results obtained by the two-pass condenser, which was installed in 1914, are equally as good if not better than those obtained with the Ingersoll-Rand modern single-pass condenser. The two-pass condenser marked "A" in Mr. Ricketts' paper does not seem to give as good results as the other condensers, but in determining engineering excellence of any particular design, comparison should be made with the best of the types representing this design rather than with the designs giving poor results.

Data published by Mr. William Weir in a paper read before the Institute of Engineers and Shipbuilders in Scotland in 1913 showed results of a number of tests on marine multi-pass pear-shaped condensers giving heat-transfer rates as high as 900 B.t.u. with vacuums of 28.5 inches Hg and better. We doubt that these results have ever been bettered by any of the modern single-pass condensers. Multipassing in heat-transfer apparatus is not and never has been an indication of inefficiency as Mr. Bancel would lead us to believe.

As Mr. Bancel states, there is no reason why the heat transfer in a two-pass and a single-pass condenser should not be the same as far as the water side of the tube is concerned, provided the tube size and water velocity are identical. This being the case, the increased heat transfer claimed for the single-pass condenser must be obtained by an improved steam distribution and more efficient air removal, with a consequent increase in heat transfer on the steam side of the tube. Mr. Bancel states that this latter result is obtained in design by shell shape, tube spacing, external coolers, and longitudinal control. He uses the conical-shaped shell with graduated tube spacing, but there is no reason why this design cannot be employed with

two-pass as well as single-pass condensers, and in fact this arrangement was used by Weir many years ago. It has also been used to a greater or less extent by many of the present-day designers of two-pass condensers. We agree with Mr. Bancel that it is a difficult matter to properly devaporize and shrink the steam-air mixture for efficient handling by the air-removal apparatus. However, we see no reason why a properly designed internal cooler should not be used. In any event the same design, whether internal or external, would apply equally as well to two-pass and single-pass condensers. The external cooler employed by Mr. Bancel has one advantage, however, in that the air-cooler surface is not included as condenser surface, thereby increasing to that extent the calculated B.t.u. transfer. The paper presented by Mr. Chatel relates that a number of two-pass condensers have from 15 to 25 per cent and over of their cooling surface in the air-cooler section. It would seem that improvement in air-cooler design would show a very marked increase in heat transfer for some of the present-day two-pass condensers.

Longitudinal control mentioned by Mr. Bancel is at best only an added complication peculiar to the single-pass design, and is automatically taken care of in the two-pass design. Also, the single-pass design with conical shell and graduated tube spacing unavoidably lends itself to the refrigeration of condensate with cold circulating water or light loads, and makes it necessary to use an external method of bypassing steam to the hotwell to overcome this difficulty.

As stated above, a design involving graduated tube spacing invites condensate refrigeration. There is, however, in the market at the present time a two-pass condenser using a modification of the graduated-tube-spacing idea which employs what might be called an adjustible central steam path or lane leading to the hotwell to overcome the refrigeration difficulty. Obviously refrigeration of the condensate cannot be controlled with varying circulating-water temperature and load without some such device. The adjustable central steam path mentioned above is regulated by baffles at the center of the condenser, these being controlled by levers from outside the shell which make it possible to regulate the amount of steam flowing to the bottom of the condenser in sufficient quantity to overcome the refrigeration.

In conclusion we would say it seems to us that Mr. Bancel's figures and arguments fail to prove that his single-pass condenser is the most efficient type, but rather point to the fact that the important considerations of shell shape, tube spacing, air cooling, and air removal should have been given more attention in the design of certain two-pass condensers.

WM. SCHWANHAUSSER.[a] This paper exploits Ingersoll-Rand single-pass condensers as the latest and best of modern surface condensers. Mr. Bancel claims that the 30,000-square-foot Ingersoll-Rand condenser at the Waterside Station of the New York Edison Company is doing the same work as a 50,000-square-foot Wheeler C. & E. condenser in the same station with the same or less water. Data presented in Mr. Ricketts' paper support this claim, but it is noted that Mr. Bancel has chosen the poorest of all the condensers listed by Mr. Ricketts to prove the contention that a "modern" surface condenser requires about half as much surface as the type usually provided. If Mr. Bancel had compared the Ingersoll-Rand unit with the old 1914 Westinghouse unit in the same station according to performance data given in Mr. Ricketts' paper, the comparison would have favored the Westinghouse unit, although the latter has been in service ten years longer. The Westinghouse two-pass condenser gives higher average yearly vacuum,

[a] Chief Engineer, Worthington Pump and Machinery Corporation, New York, N. Y. Mem. A.S.M.E.

practically the same quantity of steam condensed per square foot of cooling surface, condenses about 38 per cent more steam per gallon of circulating water, and requires less cleaning to accomplish these results.

Mr. Bancel claims that there is no essential difference between operating and test performances, and then cites one instance to prove this claim. From the information given in Mr. Ricketts' paper, it is known that the condenser selected as an example is cleaned more often than the other three condensers. Therefore the condenser in question is kept right up to test condition at all times, and so there is no reason why test and operating performances should differ. This is not, however, true in the majority of power stations, and the author's claim on this point cannot be conceded. No power station can stand the outages and costs incident to keeping condensers in test condition.

Mr. Bancel also claims that the results obtained are based on fundamental improvements in design which produce high efficiency in the utilization of surface with small expenditure of power. A comparison of units "B" and "D" in Mr. Ricketts' paper does not show that the improvements in design on surface condensers during the last twelve years have given any better results, nor that the present-day results have been obtained with any less power. Of course, condensers are smaller for the same kilowatt turbo-generator output now than a few years ago, but the greatest factor in this development has been the increase in turbine economy aided by the utilization of higher steam pressures and superheat temperatures, and the tendency toward interstage bleeding of turbines. There is no evidence that any improvements in condenser design as exemplified in the Ingersoll-Rand condenser have improved results. Certainly the comparison made by Mr. Bancel is not convincing to an engineer, and cannot be accepted as proof.

A careful study of the paper fails to disclose any real contribution to the engineering of condensers in general, nor any advance in design of condensers to improve power-plant operation or power costs. Such performance data as are given are in no way unique or new, as other condensers than the Ingersoll-Rand are doing as well or better, and this is so old a story that those concerned with these condensers have refrained from reporting results to the Society. It seems as if Mr. Bancel is evidently not familiar with these facts, and, in the opinion of many of those present, has approached dangerously close to what the Society forbids: advertising the apparatus of the company he represents.

DISCUSSION OF E. B. RICKETTS' PAPER ON SOME RESULTS OF CONDENSER OPERATION.

G. L. KOTHNY.[1] Mr. Ricketts' paper is one of those useful papers which present actual operating data and thereby point out a lesson. It deals with the results obtained with four different types of condensers, results which, as he says, will provide condenser designers and operators with food for thought.

A brief analysis of the data published would indicate that the results obtained from the old, conservatively designed two-pass condenser "D" are better than those obtained from the single-pass condenser "B," which has been widely advertised as the "ace" of modern high-efficiency surface condensers, or than those of the single-pass condenser "C," which incorporates the novel idea of subdividing a large condenser into groups of tube or tube tests—which design has been abandoned as quickly as it was adopted.

Referring, for instance, to the data published for the month of November, condenser "D" produced the highest vacuum, viz., 29.14 inches, condenser "B," 28.81 inches, and condenser "C," 28.89 inches. The ratio of circulating water to steam condensed is 75 for condenser "D," 80 for condenser "B,"

and 91.5 for condenser "C." The estimated condenser-tube friction loss is 6.6 feet in condenser "D," 6 feet in condenser "B," and 8.4 feet in condenser "C." Assuming an efficiency of 80 per cent for the circulating pump, the power input for each 100,000 pounds of steam condensed is 31.6 brake horsepower for condenser "D," 28.9 brake horsepower for condenser "B," and 48.8 brake horsepower for condenser "C."

The heat-transfer rate is 459 B.t.u. for condenser "D," 365 B.t.u. for condenser "B," and 390 B.t.u. for condenser "C."

Condenser "D" compared with condenser "B" produces 0.29 inch better vacuum with only 2.8 brake horsepower larger power input to the circulating pump per 100,000 pounds of steam condensed. The increase in vacuum will decrease the water rate of a modern turbine about 2 per cent, or increase its power output about 200 kilowatts. A net gain of 198 kilowatts will be obtained through the higher vacuum.

Similar conclusions can be drawn when considering the average performance of the four condensers over a year. Condenser "D" undoubtedly seems to be the most efficient one when considering not only the operating results but also the cost of maintenance.

During a year's operation it was washed and sandblasted only once. The single-pass condenser "B" had to be sandblasted three times and washed fifteen times, and single-pass condenser "C" had to be sandblasted three times and washed thirteen times, with no large variation in operating hours. Condensers "B" and "C" are only two years old, while condenser "D" is twelve years old. One cannot help asking how many times these condensers will have to be sandblasted and washed when they reach the age of condenser "D" in order to maintain the performance results which they are giving now. Also what the life of the tubes will be if they have to be sandblasted so frequently. Each sandblasting decreases the thickness of the tube walls.

Another point which should receive the attention of condenser operators is the number of tubes renewed per annum. Condenser "A" takes the lead with 474, condenser "C" follows with 147, condenser "D" is next with 42, while for condenser "B" so far no renewals have been reported. It would be interesting to hear from the author of this paper if these renewals have been due to water conditions or to the makes of tubes, the method of fastening them in the tube sheet, etc.

It is to be regretted that the author did not include in the published results the temperature of the condensate in the hotwell with no drains from the stage feedwater heater entering the latter.

These data are collateral to the performance results published, and therefore should not have been omitted. It is a well-known fact—and another paper dealing with condenser performance and presented at this meeting confirms it—that in the single-pass condenser the undercooling of the condensate averages higher than in two-pass condensers.

To recapitulate: The performances of the single-pass condensers as illustrated in the paper do not compare favorably with those of well-designed two-pass condensers, and their maintenance costs are high due to the necessity of frequent cleaning, which also reduces the period of availability for service.

Consideration should be given to these points by condenser operators when seeking condensers best suited to their requirements.

D. W. R. MORGAN.[4] Mr. Ricketts' paper covering weekly tests extending over a period of one year on four radically different designs of condensers

[4] Chief Engineer, Condenser Department, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa. Assoc.-Mem. A.S.M.E.

is extremely valuable from the standpoint of comparing the designs initiated over a number of years, for it has been difficult to obtain operating data on radically different designs performing under similar conditions.

For the month of September the table presented by Mr. Ricketts shows that units "A," "B," "C," and "D" condensed 5.8, 8.83, 9.0 and 9.81 pounds of steam per square foot, respectively. Analyzing the September performance further, using the water quantities indicated, and assuming a pump efficiency of 80 per cent applicable to the four units (which gives the pumps used in connection with the single-pass condenser the benefit of the doubt due to their operating at lower total head than the pumps used in connection with the two pass), indicates 288, 134, 105, and 205.5 horsepower, respectively, for condensers "A," "B," "C," and "D," all of which is set forth in Table 1.

TABLE 1. ANALYSIS OF CONDENSER DATA.

Condenser	Sq. ft. surface	No. of passes	Lb. steam condensed	Lb. steam per sq. ft.	Gal. per min.	Water velocity, ft. per sec.	B.t.u. heat transfer	Horsepower June	Horsepower July	Horsepower Sept.
"A"	50,000	2	290,000	5.8	65,000	6.5	329			288
"B"	30,000	1	265,000	8.83	57,000	6.4	588		187*	134
"C"	30,000	1	270,000	9.0	50,000	5.9	476	137†		105
"D"	25,000	2	245,000	9.8	41,000	7.3	667		104.8‡	205.5

Accepting this difference in horsepower without further investigation, one might argue that there is considerable saving in horsepower in favor of the single-pass condenser as shown by these tests. However, the condition is reversed during the month of July, which indicates 187 horsepower for condenser "B" and 104.8 horsepower for condenser "D."

Table 2 shows the pounds of steam condensed per square foot of surface, the horsepower for each month, and the average horsepower for the year. The average for condenser "B" is 144 horsepower; condenser "C," 130.7 horsepower; and condenser "D," 99.7 horsepower. A better study of the actual horsepower requirements could be made if Mr. Ricketts' paper indicated the number of hours' operation per month.

Considerable stress has been placed on the necessity of incorporating in the structure of single-pass condensers elements that will meter the steam quantity to a certain portion of the tube nest, while others advocate building in one shell what constitutes at least two or three separate condensers, each of which would operate at a different back pressure. If we expect to reach ultimate performance it can only be accomplished by directing our thoughts of design toward a structure in which equilibrium of steam distribution and absolute pressure is realized.

It is important that we exercise diligence in the design of a condenser, and it is of equal importance to correlate any excellence of design with the exhaust structure of the turbine. Obviously any attempt made to equalize the steam flow in the condenser is useless if the tube length bears no relation to the turbine exhaust. This is particularly true in the modern station where it is important that the head room be reduced to a minimum in order to minimize the station cost.

The writer is of the opinion that a large part of the tube inactivity is due to poor distribution of water to the tubes, and in this connection takes the

* Circulating 55,000 gallons per minute.
† Circulating 65,000 gallons per minute.
‡ Circulating 80,000 gallons per minute.

liberty of referring to an article appearing in the November 9, 1926, issue of "Power," entitled Condenser Study Shows Poor Water Distribution. As a matter of fact, the water boxes on this particular condenser are liberally designed and attempts have been made to obtain satisfactory distribution. The water boxes are unusually deep for the purpose of maintaining a low velocity and reducing the loss from the first to the second pass. We can only improve the hydraulic efficiency of this portion of the structure by a

TABLE 2 DATA ON HP. AND WEIGHT OF STEAM CONDENSED PER SQ. FT. OF SURFACE

Month	Gal. per min.	Gal. per min. per tube	Velocity	Hp.	Lb. steam condensed per hour	Lb. steam condensed per sq. ft.
CONDENSER "B"—30,000 Sq. Ft. Surface, 6006—7/8-in. Tubes, 21.9 Ft. Active Length						
August	57,000	9.5	6.4	134	280,000	9.34
Sept.	57,000	9.5	6.4	134	265,000	8.84
Oct.	64,000	10.66	7.16	178	290,000	9.67
Nov.	50,000	8.33	6.10	108	315,000	10.5
Dec.	52,000	8.67	5.85	104	330,000	11.0
Jan.	60,000	10.0	6.75	153	320,000	10.7
Feb.	58,000	9.68	6.51	139	325,000	10.8
March	60,000	10.00	6.75	153	327,000	10.9
April	64,000	10.66	7.14	178	325,000	10.8
May	57,000	9.50	6.40	133	325,000	10.8
June	56,000	9.34	6.30	128	295,000	9.83
July	65,000	10.83	7.30	187	275,000	9.17
Average Horsepower.....144						
CONDENSER "C"—30,000 Sq. Ft. Surface, 5694—7/8-in. Tubes, 23.3 Ft. Active Length						
Aug.	48,000	8.74	5.7	95	185,000	6.16
Sept.	50,000	8.78	5.91	105	270,000	9.00
Oct.	57,000	10.00	6.75	150	290,000	9.67
Nov.	57,000	10.00	6.75	150	310,000	10.3
Dec.	48,000	8.44	5.7	95	325,000	10.8
Jan.	46,000	8.08	5.45	84	320,000	10.7
Feb.	37,000	6.50	4.40	44	330,000	11.0
March	61,000	10.72	7.2	177	330,000	11.0
April	58,000	10.2	6.88	158	325,000	10.8
May	56,000	9.84	6.6	142	310,000	10.3
June	55,000	9.67	6.52	137	300,000	10.0
July	60,000	10.52	7.10	171	275,000	9.17
Average horsepower.....130.7						
CONDENSER "D"—25,000 Sq. Ft. Surface, 5612—1-in. Tubes, 17 Ft. 3 In. Active Length						
Aug.	33,000	11.75	5.88	114.5	205,000	8.2
Sept.	41,000	14.60	7.30	205.5	245,000	9.81
Oct.	32,000	11.40	5.70	105.0	220,000	8.18
Nov.	30,000	10.68	5.35	86.4	205,000	8.2
Dec.	29,000	10.33	5.15	76.6	205,000	8.2
Jan.	29,000	10.33	5.15	76.6	215,000	8.6
Feb.	27,000	9.62	4.80	62.3	245,000	9.8
March	31,000	11.04	5.50	93.8	240,000	9.6
April	30,000	10.68	5.35	86.7	200,000	8.0
May	28,000	9.97	5.00	70.8	190,000	7.6
June	33,000	11.75	5.88	114.5	220,000	8.8
July	32,000	11.40	5.70	104.8	185,000	7.4
Average horsepower.....99.7						

radical change in design, which will of necessity increase the cost of the structure and the space required. Here again the ultimate cannot be secured unless the circulating pumps and piping are located with the thought of improving the hydraulic efficiency.

The writer believes that the results that various people get from tests should be correlated and coördinated by such men as McAdams, Sherwood,

and Turner, who have the time and who are sufficiently interested to get the meat out of them. He believes results obtained from time to time should be submitted to such men as these, for in that way we can arrive at a better understanding of subject than we have at the present time.

As a matter of interest, the writer calls attention to the tube-sheet layout of a 22,000-square-foot single-pass condenser installed in 1917 (Figure 3), in which a low water velocity of 2.29 feet was used and a B.t.u. heat trans-

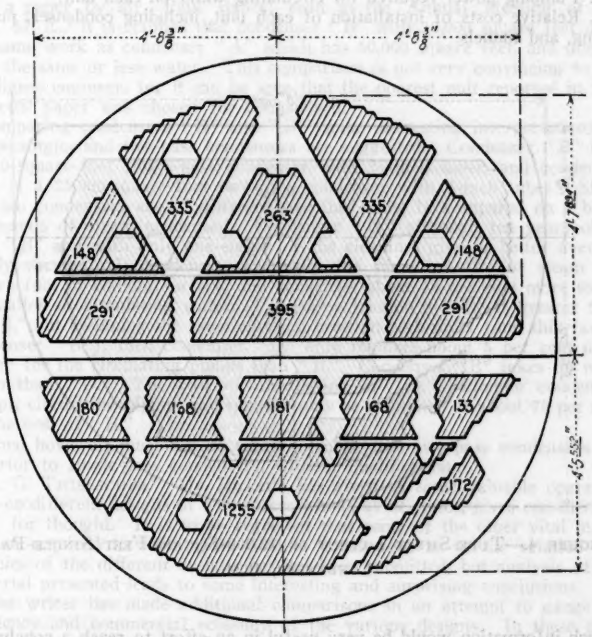

FIGURE 3.—TUBE-SHEET LAYOUT OF 22,000-SQUARE-FEET SINGLE-PASS CONDENSER, 1917.

fer of 334 obtained, which corresponds to 540 B.t.u. at 6 feet; also the tube-sheet layout of a 12,000-square-foot condenser (Figure 4), employing a water velocity of 3.88 feet per second and obsorbing 403 B.t.u., which will compare with 500 B.t.u. at 6 feet per second. This condenser was shipped in 1924.

WM. SCHWANHAUSSER.[3] This paper presents a record of commercial operating conditions covering a period of one year on four radically different designs of surface condensers, all operated under the same water conditions and under the same supervision. No claims are made for any particular

type, and the writer considers the paper a valuable engineering contribution. He does, however, feel that its value would be greatly increased if the following data were given:

1. Number of hours each unit operated each month so that truer average operating conditions may be determined.
2. Dates of installation of new tubes and number installed on each date. Condensers having greatest number installed may have been operating under handicap of reduced surface due to plugged tubes.
3. Pumping power required for circulating water on each unit.
4. Relative costs of installation of each unit, including condenser, pumps, piping, and tunnels.

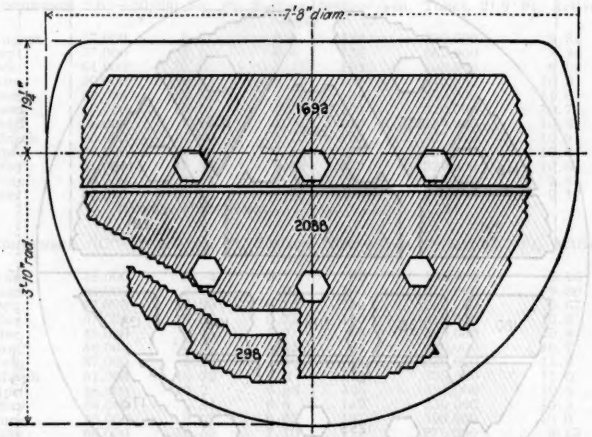

FIGURE 4.—TUBE-SHEET LAYOUT OF 12,000-SQUARE-FEET SINGLE-PASS CONDENSER, 1924.

Such information would be very useful in an effort to reach a conclusion as to the best type of condenser or the value of any one feature. In the final analysis, the decision must be based on dollars and cents; the cost of producing and maintaining a given vacuum as compared with value of the vacuum to the power plant as a whole. It is obvious that if one condensing plant were able to produce a higher vacuum than another by enough difference to add 5 per cent to the kilowatt output, then roughly it is worth $5 per kilowatt more on a total plant cost of $100 per kilowatt of capacity. Of two condensing plants substantially equal in vacuum produced, the differences in first costs, operating charges, and charges for outages must be evaluated to get a net difference which will favor one or the other.

Experience has failed to indicate any particular relation between such matters and the much-more-advertised high transfer rates. If the figures given by Mr. Ricketts prove anything, it is the fact that high vacuum is not the result of high rates of heat transfer, and of course on the above

basis of valuation only a little better vacuum would be worth to the power station several times the cost of excess tube surface over a condenser with highest heat-transfer rate and least surface.

A few years ago much publicity was given to condenser "A," and a heat-transfer coefficient of 600 was reported under conditions of cold water and high vacuum. The average annual heat-transfer coefficient of this same condenser is reported as 258 in Mr. Ricketts' paper. This shows the fallacy in judging condensers by tests or by taking a high heat-transfer coefficient as indicative of efficient and economical condensation.

In a paper by Mr. Paul Bancel read at the Annual Meeting of the A. S. M. E.,[5] it is claimed that condenser "B" with 30,000 square feet does the same work as condenser "A" which has 50,000 square feet, and does it with the same or less water. This comparison is not very convincing to the intelligent engineer, for it can be seen that the poorest unit reported in Mr. Ricketts' paper was chosen for comparison.

Comparing condensers "B" and "D," some interesting interpretations as to the single- and two-pass condensers are suggested. Condenser "B" is a 30,000-square-foot single-pass condenser with ⅞-inch tubes, and condenser "D" is a 25,000-square-foot two-pass condenser with 1-inch tubes. Since the two condensers are of different size they must be compared on a basis of pounds of steam condensed. Condenser "D," although ten years older than "B" and with only one-eighth of the cleaning, gives a better average yearly vacuum. It condenses practically the same quantity of steam per square foot of cooling surface, and condenses about 38 per cent more steam per gallon of circulating water. Condenser friction in "D" is greater than in "B," but assuming the same external head on both units (and this favors condenser "B"), then condenser "D" only requires about 5 per cent more power for the circulating pumps than "B." Condenser "B" takes up more room than "D." The cost of installation of "D," including condenser, pumps, circulating piping, and tunnels, may be estimated at about 75 per cent of the cost of "B."

This, however, must not be taken to mean that two-pass condensers are superior to single-pass condensers, without qualifications.

H. G. THIELSCHER.[6] Mr. Ricketts has given us some valuable operating data on different designs of condensers which, as he states, gives considerable food for thought. It is to be regretted that some of the other vital information required to form more definite conclusions as to the relative efficiencies of the different designs has not been submitted, but analysis of the material presented leads to some interesting and surprising conclusions.

The writer has made additional comparisons in an attempt to gauge the efficiency and commercial economy of the various designs. In these comparisons no allowance has been made for the fact that condenser "B" is favored due to not making any allowance for the separate air cooler, whose surface has not been included. Considering the ratio of condenser-shell free area to the cross-sectional area of the condenser tubes as 100 for condenser "D," we have 114 for "A," 280 for "B," and 212 for "C." Condensers "B" and "C," installed in 1924, exemplify the practice of large areas for steam flow to all the tubes, equal short paths to the air cooler, proper draining of the tubes, and uniform steam velocities, all of which factors should

[5] See p. 219.

[6] Mechanical Engineer, McClellan & Junkersfeld, New York, N. Y. Assoc.-Mem. A.S.M.E.

result in improved economy. As a result of this opening up of the steam space, we find that the index of pounds of steam condensed per hour per cubic foot of condenser volume runs as follows:

100 for condenser "D"
63 for condenser "A"
58 for condenser "B"
74 for condenser "C"

Thus, we should not expect condenser "D" to give the highest average heat transfer. If we look for the answer for this, we cannot find it in the matter of water velocities since they are as follows:

"A"—6.7 feet per second
"B"—7.3 feet per second
"C"—7.4 feet per second
"D"—7.1 feet per second

nor in the matter of pounds of steam condensed per square foot of heating surface, since the ratio, again considering "D" as 100, is:

68 for condenser "A"
112 for condenser "B"
112 for condenser "C"

nor in the matter of pounds of steam condensed per gallon of circulating water, since this ratio, again considering "D" as 100, is:

81 for condenser "A"
83 for condenser "B"
90 for condenser "C"

If we consider the circulating-water pump power, the ratio of power to pounds of steam condensed, again considering "D" as 100, is:

126 for condenser "A"
75 for condenser "B"
71 for-condenser "C"

Summing up, these comparisons indicate that condenser "D" must be lowest in first cost, and slightly higher in operating cost than "B" and "C" and lower than "A." Condenser "D" gives a higher average vacuum than any of the others, and the gain over "B" and "C" should more than offset the difference in pumping costs.

Unless Mr. Ricketts can point out valid reasons which excuse the poorer performance of the later-day condensers, the writer must conclude that much money has been wasted in the new designs, and that condenser designers must start anew with renewed vigor to keep pace with the gains in economy which have been realized in other types of power equipment.

DISCUSSION OF F. J. CHATEL'S PAPER ON STEAM-CONDENSER PRACTICE AND PERFORMANCE.

C. D. ZIMMERMAN.[7] The valuable data given in this paper and the paper by E. B. Ricketts should furnish a step toward a more standardized and more efficient design of condenser, developed by analytical methods in place of the cut-and-try methods which have been necessary so far.

[7] Production Engineer, Steam Department, Cleveland, Ohio, Electric Illuminating Company. Mem. A.S.M.E.

The object of the paper as stated is to furnish data by which the performance of several different condenser designs can be compared. These show that the recent designs have so improved in efficacy that improved results may be obtained with less surface than was used before. It is hoped that sufficient data will be available for future papers on this subject to enable readers to determine the relative value on a cost-of-operation basis of different condenser designs.

The usual method of comparison is to determine for each condenser considered total cost figures, which are plotted against the kilowatt output of the unit. The condenser having the lowest cost curve for the loads to be carried is then shown to be the most economical. Since the turbine water rate varies with the vacuum, the heat input to the turbine less the heat returned by the condenser is included in the total cost. The other factors of the total cost are the cost of the auxiliary power and the fixed charges chargeable to the condenser. In order to make these data applicable to any installation the vacuum which the condenser furnishes should be plotted against the steam or heat input to the condenser, and the pressure drop of the circulating water across the condenser should be given for different amounts of circulating water. Other factors always considered are the amount of air leakage and the relative cleanliness.

In order to predict approximately the performance of a condenser, the following equation may be of value:

$$t_v = t_i + w_s H \left(\frac{1}{2w_w} + \frac{1}{Sk}\right)$$

This equation is derived as follows, taking:

w_s = steam condensed per hour
H = heat removed per pound of steam
w_w = pounds of circulating water per hour
t_v = temperature of steam in the condenser
t_i = temperature of the inlet circulating water
t_o = temperature of outlet circulating water
t_m = arithmetical mean temperature difference
k = heat-transfer rate.

$$t_m = \frac{2t_v - t_i - t_o}{2} \quad \ldots\ldots\ldots (1)$$

$$w_s H = w_w (t_o - t_i) \quad \ldots\ldots\ldots (2)$$

$$k = \frac{w_s H}{S t_m} \quad \ldots\ldots\ldots (3)$$

From (2)

$$t_o = \frac{w_s H}{w_w} + t_i \quad \ldots\ldots\ldots (4)$$

from (1) and (3),

$$k = \frac{2w_s H}{S(2t_v - t_i - t_o)} \quad \ldots\ldots\ldots (5)$$

from (4) and (5)

$$k = \frac{2w_s h}{S\left(2t_v - t_1 - \frac{w_s H}{w_w} - t_1\right)} \quad \ldots\ldots\ldots \quad (6)$$

$$k = \frac{2w_s H}{S\left(2t_v - 2t_1 - \frac{w_s H}{w_w}\right)} \quad \ldots\ldots\ldots \quad (7)$$

and from (7)

$$t_v = t_1 + w_s H\left(\frac{1}{2w_w} + \frac{1}{Sk}\right) \quad \ldots\ldots\ldots \quad (8)$$

That is, the temperature of the exhaust steam, and therefore the condenser vacuum, can be predicted provided the factors of steam condensed, amount of circulating water, and rate of heat transfer are known. Of course the difficulty lies in determining the heat-transfer rate. Outside of the cleanliness factor and the air leakage, the heat-transfer rate depends mainly upon the penetration of steam into the condenser and upon the velocity of the circulating water. The penetration depends upon the efficacy of the design, and increases with an increase in the amount of steam condensed and decreases with an increase in the mean temperature difference. Data are available on the variation in the heat-transfer rate with variations in the amount of steam condensed, in the mean temperature difference, and in the velocity of the circulating water. With these facts in mind one may estimate the probable vacuum, or change in vacuum, for changes in amounts of steam condensed, circulating water, and surface.

Condenser experiments leading to improved conditions have been planned at the Lake Shore Station of the Cleveland Electric Illuminating Company for some time, but in order to determine the effect of each change it was necessary to know the air leakage, since an unknown variation in this factor would make it impossible to determine the effect of the changed conditions on the condenser performance. Since most of the condensers were equipped with wet vacuum pumps without means of measuring the air leakage, it was found necessary to develop a means of making this measurement. This was done by using a bypass around the valve in the air line from the condenser to the air pump. To determine the air leakage it was necessary only to close the main valve and take readings on a flow nozzle in the bypass. This apparatus has been checked on a unit having a steam-jet air ejector whereby the air discharged at atmospheric pressure was measured, and has been found to be very accurate. The calculations, method of use, and condenser-performance studies will be published at another time.

It seems to the writer that there are two other factors in connection with condenser vacuum which have not had sufficient consideration: namely,

1. Steam leaving the last blades of the turbine is at a high velocity and in a highly turbulent condition. There is evidence that this steam does not meet the condenser surface evenly but "piles up" in certain sections. It is possible that improvements can be made in the connections from the turbine to the condenser which will lead to improved condenser performance. Experiments to determine the conditions in condenser connections at Lake Shore have been planned for some time, but have been postponed due to the press of more urgent work.

2. Steam leaving the last turbine blades enters an open space without streamlining, which is analogous to using an orifice in places where an efficient nozzle should be used. This sudden increase in the area of the passage causes violent eddies and turbulence as the steam changes from a high velocity on leaving the last blades to a lower velocity on entering the condenser shell. Due to this turbulence this change in state is accompanied by an appreciable increase in entropy and a consequent loss in the availability of the energy which, for the same vacuum at the top of the condenser, might be utilized in the turbine itself.

GENERAL ORAL DISCUSSION.

D. W. R. Morgan,[4] speaking on Mr. Bancel's paper, said that as far as he knew, it took just as many B.t.u. today as it did ten years ago to raise a pound of water one degree. Our goal in condenser design should be to minimize the drop through the condenser and our thought should be directed along that line.

Mr. Bancel had made a very good point in suggesting that we use a method that might be recognized by every one in defining airpump capacity. There was no reason why the Condenser Committee of the Society should not get together and define it. It should be defined by stating a definite quantity of air leakage, and the definition should take into consideration the temperature of the mixture at the air offtake and the amount of condensable vapor that might be carried over at that temperature.

Mr. Bancel referred to an N. E. L. A. publication as to the temperature in a two-pass condenser. Mr. Morgan believed, however, that there were sufficient data of more recent date that indicated clearly in the modern design the two-pass condenser would obtain approximately the same temperature rise in the first half as in the second.

G. L. Kothny,[1] referring to a remark made by Mr. Morgan in regard to the air-handling capacity of air pumps for different sizes of condenser installations, said that not only the actual capacity as far as free air was concerned but also the temperature of the air-vapor mixture should be considered, and that this had already been done by the United States Navy at Annapolis.

George A. Orrok[8] said that the three condenser papers were very interesting to him because they illustrated a point which he thought was frequently forgotten: namely, that no condenser in commercial service was ever really tested for heat transfer. All that the test meant was that what heat was there was transferred. In other words, it had never been tested up to the limit of the condenser surface.

When a heat-transfer test on a tube was made, all of the heat was supplied to the wall that could be transferred through the tube. When such a condenser test or record was made, one got just the transfer of the heat that was there. At 300 B.t.u. a transfer did not mean anything. The tubes were good for 1400 to 2000 B.t,u. if that amount of heat was there to be transferred. But tests were never made that way, and no operating man would ever buy a condenser with a small enough surface so that he could actually see what that surface could do.

Leo J. Levit,[9] referring to a statement made by Mr. Morgan, said that while it took as many B.t.u. today as ten years ago, at that time turbines were not bled as extensively as now. There was a difference between B.t.u. at high and low heat levels.

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[8] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.
[9] Power Plant Department, United Electric Light & Power Co., New York, N. Y. Jun. A.S.M.E.

Abbott Allen[10] said that the operating man was interested in getting just as high a vacuum as he could. He wanted a condenser that would use as little circulating water as possible and was easy to keep clean, and once every five years, perhaps, he thought about the cost of tube replacements.

The viewpoint of the man who designed condensers was or should be that he would like to get the best possible overall performance out of a minimum factory cost. That of the man who designed the power station was or should be that he would like to get the minimum overall cost of power at the station. This overall cost was made up of two components: namely, the operating cost and the fixed charges.

When one went into a further analysis of this latter viewpoint, which by and large should be the viewpoint of every one who was interested in condenser design and performance, he would find the matter was rather involved. In fact, nine out of ten, whether operating men or designing engineers, had found the problem so involved that they had thrown up their hands, and most of their decisions on the selection of type or the proportioning of surface and water quantities and number of passes had been made on the basis of pure guesswork, superstitions, preconceived ideas, and extraneous matter.

Mr. Allen had found that it was possible to make an analysis which would enable one to weigh the relative merits of condensers of different types or of different proportions. It was necessary, however, before making that analysis to form an accurate idea of loading conditions on the generating unit during its active life—which ought to be taken as at least fifteen or twenty years—to find out, corresponding to, say, 100 per cent, 70 per cent, and 40 per cent of the maximum throttle flow, for which the throttle was designed, what the number of hours of operation would be during each of the twelve months of the year.

The second thing as bearing on load conditions was that of accurately determining or determining by guesswork the date on which maximum demand would probably occur.

The next thing was to determine what the average circulating-water temperature would be during each month of the year. There must also be accurate curves showing the bleeding water rates of the turbine, corresponding to 100, 70, and 40 per cent of maximum throttle flow over a range of back pressures from a half-inch to, say, 3 inches; accurate curves of B.t.u. rejected to the condenser over a range of back pressures corresponding to, say, three different throttle flows; and an accurate knowledge corresponding to, say, these three different throttle flows as to what back pressures a given condenser would produce under day-in-and-day-out conditions; not test performance, but day-in-and-day-out performance, corresponding to different throttle flows. Further, one must be able to guess accurately the cost of tube replacements and the frequency with which they must be made.

There had been such an advance in condenser design that a salesman attempting to sell the equipment of eight years ago at the present time would almost have to give it away in order that the yearly costs would be equal to those of the condensers available today.

P. E. Reynolds[9] agreed with Messrs. Bancel and Kothny that some unit should be developed to determine the air-handling capacity of an air pump, that some unit should be designed which had a relation to the actual operation of the air pump. He was a member of the Test Code Committee, and would like to ask Mr. Kothny how the temperature of the air mixture was to be determined. He had had some little experience in the matter, and

[10] Assistant Mechanical Engineer, Stone & Webster, Inc., Boston, Mass. Mem. A.S.M.E.

knew it was very difficult to measure the temperature of the air mixture, particularly when the absolute pressures were 1 inch or below. The density of the vapor at that stage or the specific volume was rather high, and it had been Mr. Reynolds' experience that it was very difficult to obtain a temperature which really represented the temperature of the mixture.

Paul Bancel said that a comparison of condensers "B" and "C," both 30,000-square-foot single-pass but of radically different designs, was of interest. In making such a comparison, the influence of small differences in vacuum on the true value of condensing machinery should be borne in mind. One-tenth of an inch of vacuum on the turbines under consideration was worth 2000 pounds of steam per hour, which capitalized to $35,000 of first cost on the basis of steam being worth 35 cents per 1000 pounds and approximately 6500 hours of operation with fixed charges at 12½ per cent. With other figures, this might be lower or higher.

It was sufficiently close to set the figure at $30,000, thus giving one-tenth of an inch of vacuum as being worth a dollar per square foot of condenser. And from this it was apparent that it only took a few tenths of an inch of vacuum to be the equivalent of the entire cost of the condenser. Putting this another way, a condenser which was inferior by two- or three-tenths of an inch was a 100 per cent failure and would have to be given without cost to the user. On the other hand, a tenth of an inch of vacuum was a small difference from a practical operating point of view based on single or superfiicial observations. In this case, however, the averages of a whole month's or year's operation could be used for conclusive comparisons, as they represented an actual measure of differences in coal burned.

Taking the average performance as given by Mr. Ricketts for condensers "B" and "C," it would be seen that these were represented by the coefficients of 432 and 392 for the two condensers over the whole year. On the basis of 300,000 pounds of steam, this difference represented approximately 0.1 inch of vacuum, which was shown above as the equivalent of a large part of the first cost of either condenser. To this also should be added any difference in jet steam consumption.

The records for May and July, that was, the month previous to sandblasting and the month after sandblasting, for both units, showed coefficients in the ratio of 121 and 122 per cent in favor of condenser "B." These coefficients translated into more nearly two-tenths of an inch difference in vacuum. The differences in actual vacuums reported in Table 1 checked this for July (0.18) when the reported loads were the same, but for May the load on "B" was higher than on "C," calling for correction to the reported vacuums. It was obvious that on the basis of the actual operating results, the difference in true value of the two condensers was a very large sum.

Mr. Bancel had made a comparison with the single-pass condensers at the Detroit Edison Co. The accompanying chart, Figure 5, at the right showed the condenser performance of the two condensers of Figure 6 and Figure 8 in the author's paper. To his mind this was a very simple way of showing condenser performance. He had also plotted the four test points given in Mr. Chatel's paper for the latest single-pass condenser of the Detroit Edison, with just short lines pointing down toward zero for those four points. Those four points should be on separate lines, because all were on different water quantities.

Mr. Bancel had taken an average of tests 58 and 65 at 6.3 pounds per square feet load and 20 degrees, which was a typical temperature difference. On the Ingersoll-Rand there was an 11-pound load to the square foot, taking an average of these two condensers, one of more recent design, than the other and in a different plant. The temperature rise was 9.7 degrees average,

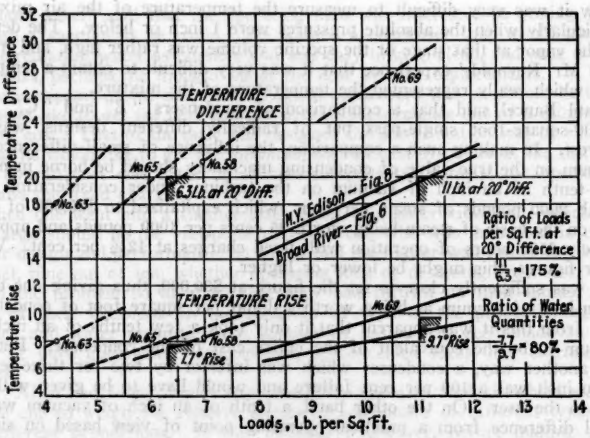


FIGURE 5.—COMPARISON OF INGERSOLL-RAND SINGLE-PASS CONDENSERS WITH SINGLE-PASS CONDENSER AT DETROIT EDISON FROM MR. CHATEL'S PAPER.

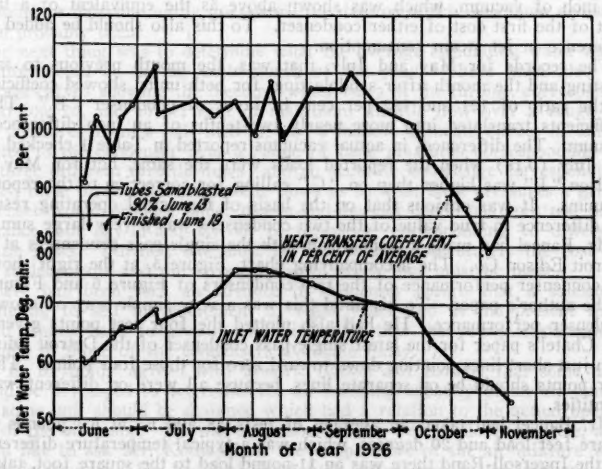


FIGURE 6.—RESULTS OF 24 TESTS OF INGERSOLL-RAND CONDENSER AT WATERSIDE.

and to the left it was about 7.7 degrees. So that the ratio of the loads per square foot was 175 per cent, and the ratio of water quantities, 80 per cent. There was, he thought, a very marked difference in the actual efficiencies of different designs of condensers.

AUTHORS' CLOSURES.

Mr. Ricketts in closing said that he thought Mr. Kothny had misunderstood the tube-renewal figures given in his paper. Their only significance was that if 2000 tubes were renewed during the year, it was evidence that during part of that time at least a considerable number of tubes had been plugged and were not in service. That fact had been put in to show that the condenser had not had the full amount of surface in service during the

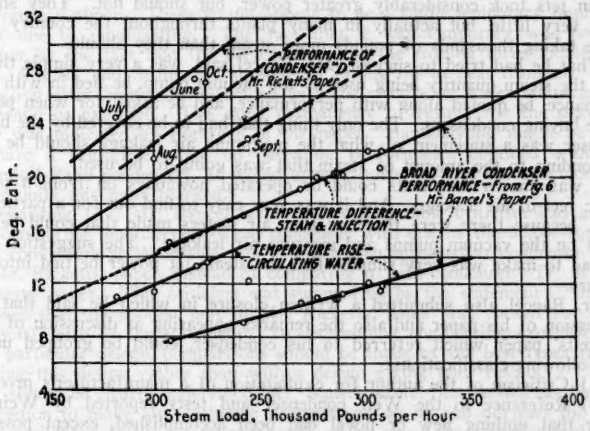


FIGURE 7.—COMPARISON OF PERFORMANCE OF INGERSOLL-RAND BROAD RIVER CONDENSER WITH THAT OF CONDENSER "D" OF MR. RICKETTS' PAPER.

whole of the year. It had no significance as to repair costs, because his company's experience was that about once every six or seven years it became necessary to renew their condenser tubes. It would be noted that condenser "A" was about six years old during the period considered, and it was about time to get a new set of tubes.

Replying to Mr. Morgan, he stated the hours of operation were given in the table in the paper, and that in the curve sheets he would find the load on the condensers.

Mr. Chatel, in closing, said that the company he represented had gone to one-pass condensers in the latest installations; that did not necessarily mean that the one-pass condenser was superior to the two-pass condenser, everything considered. He believed, however, that the performance of these condensers, even though they might use more water than was necessary with the two-pass condenser, warranted their installation.

It had been shown that some condensers had a lower ratio of water quantity to the load. Some condensers had their surfaces reduced to such an extent that although the heat transfers were high, the safety factors were low and they required cleaning too frequently. The turbines had to be shut down oftener, and it was the policy of his company to install enough surface to result in safety factors which would not necessitate crippling the units too often.

Mr. Bancel in closing said that he did not think the matter of the vacuum pump, air temperature, etc., was so complicated as it might seem to some. It was a peculiar thing that one would see condenser performance quoted with elaborate calculations and conclusions, while the power for the vacuum pump was forgotten. This was proper enough ten or fifteen years ago when reciprocating vacuum pumps were used. The reciprocating vacuum pump took practically no power, the load being almost all friction. However, steam jets took considerably greater power, but should not. They should take very little, but actually in many plants throughout the country they were taking thousands of pounds more steam than they should.

What he had tried to suggest, Mr. Bancel said, was a very simple thing: that the steam quantity being used by the vacuum pump, be tied in with performance, be quoted along with performance, and be asked for when people were buying condensers. The only thing that had to be covered by the buyer or user was a statement of what the maximum air leakage should be corresponding to the amount of steam that was going to be used.

It was known that units could be operated nowadays on from 3 to 10 cubic feet of air leakage. And it was very easy to find out for a particular case, because there were three or four air meters made that could be put right on the vacuum pumps and measure air leakage. The suggestion that he had to make was very simple—that the steam-jet power be tied into the picture.

Mr. Bancel also submitted a written closure in which he said that the discussion of his paper and also the remarks appearing as discussion of Mr. Ricketts' paper which referred to his condenser, could be grouped under the following classifications:

(1) Criticism of the author for exploitation of a manufacturer's product.

(2) Reference to the Weir condenser and tests reported by Weir to show that nothing new or novel had been accomplished, except possibly the change from multiple pass to single pass. Under this heading also came reference by Mr. Reynolds to a test of one of his condensers.

(3) Criticism of Mr. Bancel's claim to having accomplished at New York Edison with 30,000 square feet the work previously accomplished with 50,000 square feet by pointing out the fact that the tubes of the 30,000-square-foot condenser had been sandblasted three times.

(4) A repeated reference to condenser "D" in Mr. Ricketts' paper which, though five years older than the other two-pass condenser, showed superior results and should be compared with the author's condenser at New York Edison.

Taking these points in their order, Mr. Bancel wrote, the first might be readily disposed of by pointing out that he had been requested to prepare a paper by the Central Station Papers Committee, Power Division, and that this paper had been submitted to the Committee for the usual approval.

Regarding (2), it had been pointed out that the heart-shaped shell was nothing new and had been used many years ago by Weir. As a matter of record, the heart-shaped shell had been used before Weir's time by another manufacturer—almost forty years ago. It was also interesting to note that air preheating and bleed heating, both subjects of frequent discussion nowa-

days, had also been used thirty and forty years ago. From the test performances quoted from Weir's paper, also the test performances quoted by Mr. Reynolds, it might be possible to draw some interesting inferences. However, the purpose of his paper, Mr. Bancel wrote, had been to present operating results and not test results. While a gratifying amount of discussion and criticism had been advanced by various members, no operating data had so far been presented.

As to point (3), a claim that his 30,000-square-foot condenser had done the work of the 50,000-square-foot condenser of a conventional type, based on operation favored by frequent sandblastings of his condenser, would be misleading to say the least. His condenser had been sandblasted twice in the summer of 1925, around the start of Mr. Ricketts' cycle of performance, and the third time in June of 1926. Starting at this time a weekly test, independent of any results of operation, had been run and Mr. Ricketts had kindly given Mr. Bancel permission to present a summary of the results in Figure 6 above. There would appear to be a slight temporary improvement due to sandblasting. The fact that it was temporary was proved further by the fact that the performance for the ensuing three months of July, August, and September showed no falling off in heat transfer. The reduction after September coincided with the circulating-water temperature, as would be expected. The same permanency of results for the previous year was indicated by Figure 8 in the paper. Sandblasting might be said to have a negligible effect on operating performance obtained with the Ingersoll-Rand condenser. It would appear, however, that the effect on the other units was more marked.

Regarding (4), Mr. Bancel wrote that considerable criticism had arisen for comparing the Ingersoll-Rand condenser with one of the 50,000-square-foot condensers at Waterside No. 1, which also appeared as condenser "A" in Mr. Ricketts' paper, instead of comparing it with condenser "D" of Mr. Ricketts' paper. This criticism was summed up by the statement: "It can be seen that the poorest unit in Mr. Ricketts' paper was chosen for comparison;" also by the statement: "In determining engineering excellence of any particular design, comparisons should be made of the best types representing this design rather than with a design giving poor results."

It should be clearly understood, Mr. Bancel wrote, that he had not been in a position to make any authentic comparisons, as he had not had the data now available. In the second paragraph of his paper the assertion had been made that the 30,000-square-foot Ingersoll-Rand condenser had done the work of the 50,000-square-foot conventional condensers, this claim being made on the basis of personal observations from time to time. In Waterside No. 1 there were four principal units, two 35,000-kilowatt machines with 50,000-square-foot condensers, and the two newer 35,000-kilowatt machines with 30,000-square-foot condensers. The turbines were similar in size and appearance and were usually operating at about the same output, the four units carrying a base load. Comparisons of performance of all four units were natural and largely a matter of comparing vacuums, which had been commonly done by visitors and by the station operating force. During the first six mnoths to a year after installation Mr. Bancel had frequently been told by engineers that they had heard that the new condenser operated lower than the two 50,000-square-foot machines, the difference quoted being as high as $\frac{5}{8}$ inch. The data now made public by Mr. Ricketts on one of these 50,000-square-foot units, which was presumably the better of the two, bore out Mr. Bancel's claims. Criticism advanced now for comparing the new condenser with units which previously had been widely used by engineers for comparison when it had been believed that they were superior, seemed to need no furthr comment.

The data now available on condenser "D," the 25,000-square-foot two-pass condenser installed in 1914, opened up another possibility for comparison. This unit was in Waterside No. 2 on a different-size turbine and in another house, so that personal comparison naturally had never been made. There were a great many other condensers in this house, and Mr. Ricketts had doubtlessly selected the one showing the very best results for presentation in his paper. Mr. Bancel believed that he was equally justified in making comparisons with any unit he might select, provided he presented authentic data, as, for example, for the 25,000-square-foot condenser at Parr, S. C., covered by Figures 5 and 6 of his paper. Both condensers had 25,000 square feet and the comparative results were shown by Figure 7 above. It would be noted that the results for "D" were highly erratic. Taking the best month for condenser "D," September, 1925, when the tubes had been sand-blasted and the heat transfer from Table 1 of Mr. Ricketts' paper jumped about 50 per cent, the chart showed loads of 245,000 pounds vs. 325,000 pounds, or a ratio of 132 per cent in favor of the Broad River condenser. Again, taking the month of July, 1926, for condenser "D," the Ingersoll-Rand Condenser carried 345,000 pounds as against 182,000 pounds, or a ratio of 190 per cent. When doing this the temperature rise of the water was 13.7 degrees as against 11 degrees indicating 80 per cent of the water per pound of steam condensed. The average performance for the five months was represented by the dotted line; at 280,000 pounds the steam temperature with condenser "D" was 12 degrees hotter than with the Ingersoll-Rand condenser. This differential in temperature was fully ½ inch vacuum and represented a loss capitalizing at about twice the cost of a condenser. Therefore, if condenser "D" of Mr. Ricketts' paper had been installed at Broad River, it would have been necessary to give it without cost and also endow the customer with a sum equal to its cost. Even taking September results for condenser "D," which were obviously far superior to the average, the vacuum at 245,000 pounds (Waterside load) was 5.6 degrees high or 0.3 inch low; at 300,000 pounds (Board River load) the vacuum temperature was 6.8 degrees high or 0.4 inch low; these differences when capitalized were more than the equivalent of the entire cost of the condenser.

A few further remarks which did not come within the four classifications mentioned above, might be answered briefly. Mr. Kothny criticized Mr. Bancel's curves for tube friction, also the use of the square-root law for the change of heat-transfer coefficient. The friction curves based on Saph and Schoder had been checked some years ago in the Ingersoll-Rand laboratory and found to be safe. Mr. Bancel wrote that he was glad to receive criticism as to their consistency as between 1-inch tubes and the smaller sizes, and this would be rechecked. As regarded the use of the square-root law for heat-transfer coefficient, he had considered this as a widely accepted and widely used relationship.

Mr. Kothny presented a photograph of a Weir condenser with support sheets extending up into the steam space. Mr. Bancel wrote, however, that his remarks about longitudinal control had referred to extension of the support sheets down to the bottom of the condenser. Mr. Kothny appeared concerned about leakage of steam through the holes in the support sheets. As to this, one of Mr. Bancel's earlier inventions on file with his legal department showed an arrangement for sealing the holes with water, but this was now known to be unnecessary.

Mr. Reynolds suggested that the important considerations of shell shape, tube spacing, air cooling and air removal should have been given more consideration in design of certain two-pass condensers. It had been his aim, in the last part of his paper, Mr. Bancel wrote, to point out that these

features were fundamentally important with any condenser, no matter how many passes it had. His company had built a number of multiple-pass condensers and employed the same features of design. They were, however, of particular interest and importance with single-pass condensers because single-passing put a premium on good design. Mr. Morgan's point that condensate temperature was of just as much importance today as ever had been answered by Mr. Levit. Readers were also referred to Trans. A. S. M. E., vol. 45, p. 733.

Mr. Alden pointed out the necessity of careful investigation when selecting the size and type of a condenser, and made the statement that the present-day condenser was far in advance of the design of eight years ago. It was Mr. Bancel's conclusion, based on his calculations and also careful studies made by others, that the single-pass condenser was indicated for the majority of conditions in the majority of plants. It was believed that most of the new turbines now building for leading stations were being fitted with single-pass condensers.—"Mechanical Engineering," March, 1927.

EROSION OF SURFACE-CONDENSER TUBES.*

By The Hon. Sir Charles A. Parsons, K.C.B., D.Sc., F.R.S.

The problem of the failure of the tubes of surface condensers is so well known that it seems only necessary here to allude to a few of the salient facts bearing upon the question. Whilst no adequate explanation has as yet been assigned for such failure, many attempts have been made to overcome it. On the theory that it is caused by electrolytic corrosion, counter-electromotive force has been applied to neutralize the action; on the theory that it arises from chemical action, such means as coating the inside of the tubes with a bituminous paint or with a protective scale of oxide have been tried. None of these measures has proved to be more than palliative in its effects, and sometimes not even that. In a few cases more expensive metals are being adopted for the tubes, such as cupro-nickel or Monel metal. For many years bell-shaped mouths have been fitted to the inlet ends of condenser tubes to produce a smooth stream-line flow into the tubes,† and this is claimed to prevent the formation of a *vena contracta* by the sharp square entrance to the tube and the liberation of oxygen and other occluded gases from the water which cause corrosive action at the tube entrance.‡ In most cases of failure it has been observed that certain of the tubes are eroded, corroded, pitted, and eventually holed, but always, or nearly always, from the water side, that is to say, from the inside surface. The action is usually confined to tubes in certain regions of the condenser tube-plate, and if such tubes are replaced from the common batch they have usually failed just as quickly, whilst in other positions in the tube-plate the original tubes have remained immune, so that there appear to be distinctive areas in which the conditions favor corrosion or erosion. Further, it appears that the action is greatest at the entry ends of the tubes, frequently extending only a few feet from the mouth. These facts seem to suggest that the cause may be

* Paper read before the Institution of Naval Architects, London, on April 6, 1927.

† Vide P. C. Parker's remarks in the discussion on Dr. Bengough's paper, Transactions of the North-East Coast Institution of Engineers and Shipbuilders. Vol. xi, Part 2 ,1928.

‡ Vide Serial Report of the Prime Movers Committee, 1926-7. The National Electric Light Association of New York, U. S. A.

hydro-dynamical rather than electrical or chemical, and, in the light of the experiments now described, seem to indicate that these areas of erosion or corrosion are associated with regions of turbulent motion of the water in the water box.

The object of the present paper is to give an account of some experimental investigations which have been designed to throw light upon the actual conditions attending such regional disturbances in the water box, and the manner in which they affect the character of the flow through the tubes in their vicinity, also to investigate the possibility that the pitting of condenser tubes may in reality be due to water-hammer of collapsing vortices, which is known to be a potent cause of erosion of screw propellers and of the impellers of centrifugal pumps and water turbines. A paper was read before the Institution in 1919 by the present author and Mr. Stanley S. Cook on the "Cause of Erosion of Propellers."§ The paper recounted the investigations carried out by the Propeller Erosion or Corrosion Committee of the Board of Invention and Research. In that paper, experiments with a water-hammer cone were described, in which it was shown that by concentrated collapse of a vacuous cavity in water under atmospheric pressure small holes could be punched through brass discs, demonstrating the production by this means of shock pressures as high as 150 tons per square inch. The conclusions of the Committee were that the powerful and erratic erosion experienced with propellers was due to vacuous cavities in the water arising from eddies caused by struts or bossing, or by excessive slip of the propeller itself, which cavities, collapsing on the surface of the propeller blade, caused repeated blows and erosion of the metal.

The first of the present experiments was designed so that the behavior of the water in the water box, and also in the tubes, could be observed and, if desired, photographed. The apparatus consists of a small wooden water box, the front being of plate glass, with a brass tube-plate for ten tubes. To facilitate observation, the first 3 feet of the four tubes of the central vertical row were made of glass also. The tubes of the side rows were omitted, and instead retarding plugs were fitted giving the same discharge as a 15-foot length of condenser tube. The discharge from these side holes was led into a separate chamber immediately behind the tube plate. The four central glass tubes passed through the back of this chamber, and were joined at their ends, by clamped rubber-tube couplings, to brass tubes 12 feet long, making a total tube length of 15 feet. They were also surrounded by a rectangular water tank with plate glass sides so that, the refractive index of glass and water being nearly the same, the water inside the tube could be seen without appreciable refraction. The discharge from the long tubes was taken into a measuring tank with a glass front, so that the trajectory of the water issuing from the tubes could be observed. Closely fitting between the front plate glass of the water box and the tube plate, an inner and movable water box of circular shape was fitted, into which the water was led tangentially through a socket fitting. This inner water box was moved by a lever passed through a stuffing box, guides and stops controlling its motion, so that it could be moved up and down across the face of the tube plate. An air vessel on the water supply to the box maintained a steady pressure of about 10 pounds per square inch. With water entering the water box at a speed of 8 feet per second, which was also the speed of the water through the tubes, a vortex filament was observable near the center of the circular inner box, extending across from the glass plate to the tube plate. By moving the box, this vortex could be

§ See Engineering, vol. cvii, pages 501 and 515.

brought over the mouths of tubes Nos. 2 and 3 of the four central tubes, the tubes No. 1 and No. 4, top and bottom, being only presented to the outer zones of the vortex filament.

The following results were observed when a vortex filament was, in this way, brought over the mouth of a tube, through which, up to that moment, the water had been flowing with a speed of 8 feet per second.

1. The vortex was drawn into the mouth of the tube.

2. The flow in that tube was partially arrested, the discharge being reduced in a fraction of a second to about one-fifth of its previous quantity.

3. The vacuum at the core of the vortex brought over the tube mouth was momentarily intensified by the arrest of momentum of the water already in the tube.

4. A vacuous thread extended momentarily into the tube for about half the length of the glass portion, and then collapsed irregularly at various points.

5. Silvery looking fringes appeared momentarily on the entry edges of all the four tubes, whether under or near the vortex.

6. It was observed that, at the instant when the vortex was brought over the tubes 2 and 3, it developed a momentary bifurcation into the neighboring tubes 1 and 4, the bifurcation being of such short duration as to be difficult to see.

The apparatus was next modified for continuous working, in order to ascertain if pitting of the tubes could actually be produced in this manner. The glass tubes were replaced by tubes of brass, continuous for the first 12 feet of their length, and then extended to 15 feet by attaching 3 feet lengths of similar brass tube. A solenoid, with a make-and-break relay, was fitted to operate the lever to oscillate the inner water box, at a rate of about 15 cycles per minute, and a motor-driven centrifugal pump and drain tank maintained a continuous circulation, the system being filled with brackish water from the river Tyne. The solenoid-operated lever was adjusted, by guides and stops, so as to bring the vortex filament alternately over the mouths of the second and third of the four central tubes, so that a comparison of these tubes afterwards with the first and fourth, which were not subjected to the action of the central part of the vortex filament, would enable any difference in effect to be definitely attributed to this action. The apparatus was run for 1450 hours, or about two months' continuous work, after which the four tubes were taken out and the first 3 feet of each sawed in two for examination. Pitting was observed at the entry of all the four tubes extending from about ⅛ inch to a few inches along the several tubes; beyond this the tubes were apparently unaffected. In one of the tubes, several pits penetrated about half-way through the thickness of the tube. There appeared to be very little difference in the amount of pitting in the two tubes subjected to the vortex filament, and those subjected only to the rapid transverse flow of water across their mouths. The experiments will, however, be continued with varied conditions, including transfer of heat across the tubes to the water as in ordinary condenser practice, and it is hoped that these may throw further light upon the subject, and produce experimentally the erosion of the tubes in the central portions of their length.

In order to investigate the character of the regional disturbances occurring in a water box of usual type, a further experiment was made. A model water box was constructed, about a quarter full size of an actual condenser box, with a tube plate for 100 tubes of ⅝-inch diameter. These tubes were bent and spread out in groups, so that the discharge from each tube could be observed. The water box was fitted with a glass front. Town water was used, which was circulated by a centrifugal pump. The velocity in the tubes was about the same as in an ordinary condenser, *i.e.*, about 8 feet per

second. The tubes being of different lengths, retarders were fitted near their mouths to give a discharge equivalent to tubes of 15 feet length, and to maintain the standard velocity in all the tubes. Regions of strong turbulence were distinctly observed in the water box, and the vortex centers were seen to move about from tube to tube over certain areas. The discharge from some of the tubes was found to show sudden variation; this variation corresponded, as regards the tube positions, with the vortex motion in the box, although the sudden drop of velocity was only about one-half that

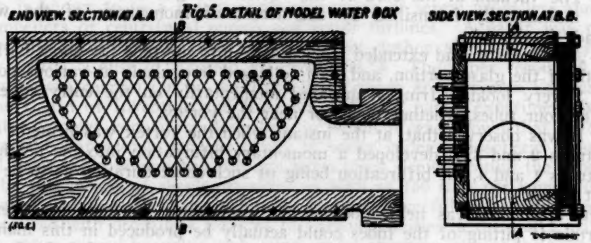


Fig. 5. DETAIL OF MODEL WATER BOX

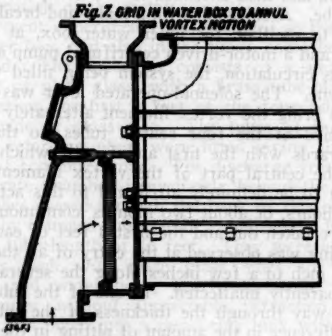


Fig. 7. GRID IN WATER BOX TO ANNUL VORTEX MOTION

observed in the previous experiments with the circular water box. It must be borne in mind, however, that the vortices, in relation to the size of tubes, would only be of about one-quarter the size they would be in an average full-sized condenser, and also smaller than the single vortex in the circular water-box experiments.

In conclusion, it is suggested that the experiments, as far as they have been carried out, seem to lead towards the view that one of the chief causes of the erosion of condenser tubes may lie in the turbulent motion of the water in the water box of the condenser, and that this cause could be mitigated by enlarging the area of the inlet passages so as to reduce the velocity of the water as much as possible before entering the box, or entirely removed by arranging in the water box a grid of suitable mesh and with passages of

depth somewhat greater than their width. Figure 1 shows such a proposal. By the passage of the water through this grid, turbulence is practically destroyed. Grids of this kind have been tried in conjunction with the experimental apparatus and it is found that with such a grid, when the whirl chamber is moved across the tube plate exactly as before, the velocity at the discharge end of the tubes remains perfectly steady, and no vortex motion is visible in the box either behind or in front of the grid.—"Engineering," April 8, 1927.

AUXILIARY SUPERCOOLING IN SURFACE CONDENSERS.

The function of a condenser is to condense the exhaust steam from a prime mover, and thus to maintain a low exhaust pressure. To do this efficiently it should (1) utilize the minimum cooling surface with minimum cooling water and maximum heat interchange between steam and cooling water; (2) discharge the condensate at as high temperature as possible, as determined by the vacuum maintained; and (3) cool the residual steam-air mixture to a low temperature before its removal by the vacuum pump.

A. J. Nicholas in "Power" points out that item 1 is the most important in condenser design, and is governed by choosing the proper size, number, and spacing of tubes and shape of chamber. The number and size of tubes give the rate of flow of the cooling water. The spacing of tubes and chamber shapes determine the rate of flow of the exhaust steam and air in their downward course through the condenser. The relation of these two determines the conditions existing in the exhaust chamber. The velocity of the steam and air should be kept uniform and constant in their travel from the top to the bottom of the condenser. This makes each tube effective in the removal of heat from the steam and prevents stratification of the steam-air mixture, which is the cause of hot spots and non-uniform temperature in the condenser, and thus makes some tubes less effective than others.

Item 2 is desirable in that it reduces the heating of the condensate when it is used as boiler feed. Item 3 is desirable because the capacity of the air-removal pump required to serve the condenser is consederably reduced by cooling the steam-air mixture entering the pump suction.

CONDENSATE V. STEAM-AIR TEMPERATURES.

Unfortunately, it is impossible to realize a high condensate temperature and a cool residual steam-air mixture in the same shell. If in a single shell, without baffling or other separation of parts, the cooling is carried to a low temperature so as to realize item 3, then the condensate temperature will be low, and item 2 is sacrificed. If, on the other hand, hot condensate is obtained, the steam-air mixture will be at the same high temperature, and a larger vacuum pump will be necessary.

The tubes in the main body of the condenser should extract the major portion of heat from the steam, leaving hot condensate at the bottom for use in the boilers. The minor portion of heat should be handled by an auxiliary cooler specially built to handle the residual steam-air mixture rich in air. Such a condenser plant is shown in the diagram of Figure 1. With a separate cooler it is possible to have hot condensate and a cool steam-air mixture before it is withdrawn by the vacuum pump, and so to use a vacuum pump of small dimensions and still maintain a high vacuum in the condenser.

To show this concretely, suppose that the condenser plant of Figure 1 is maintaining at the steam inlet an absolute pressure of 2 inches of mercury.

Thermocouples at various points in the condenser shell will show no appreciable drop in temperature until more than 99 per cent of the steam has been condensed. Throughout the main condenser shell the temperature will be substantially 101 degrees F., but at the point of departure of the residual steam-air mixture, where there remains only a trace of steam, the temperature drops rapidly as condensation proceeds. Suppose that at this point the temperature has fallen to 96 degrees F. In a simple open condenser shell the condensate temperature will also be about 96 degrees F.

While passing through the external cooler shown in the diagram of Figure 1 the steam-air mixture is cooled and devaporized. The typical curve of Figure 2 shows what takes place. The portion A, B of the steam is condensed by the main condenser tubes, and the portion B, C by the auxiliary supercooler. Thermocouples located in the cooler will show a drop in temperature as the steam-air mixture approaches the vacuum-pump suction connection.

If no cooler were present the residual steam-air mixture would be drawn off at the same temperature as the condensate. The partial pressure of the steam in this mixture, as found in the steam table, would be 1.7 inches of mercury, and that of the air would be 0.3 inch of mercury, the sum being equal to 2 inches, the absolute total pressure assumed. The volume of this air at the condition of withdrawal from the condenser would be approximately $30 \div 0.3 = 100$ times the volume of free air at atmospheric pressure.

Suppose, on the other hand, that the mixture is passed through a cooler and is cooled to 90 degrees F. before withdrawal by the vacuum pump. The partial pressure of the steam at this temperature would be 1.4 inches of mercury, and that due to the air would necessarily be 0.6 inch of mercury, since the sum must equal the absolute pressure of 2 inches (friction having only a small effect on pressure). The volume of this air at the condition of withdrawal from the cooler would be $30 \div 0.6$ inch $= 50$ times the volume of free air at atmospheric pressure. It is evident, then, that by reducing the temperature of the steam-air mixture 6 degrees before withdrawal by the vacuum pump, the volume of air that the pump must handle may be reduced to half. This means that a given pump would handle twice the weight of air for the same effective displacement.

EFFECT OF SUPERCOOLING ON THE VOLUMETRIC EFFICIENCY OF THE VACUUM PUMP.

Figure 3 shows how supercooling affects the volumes of air which the pump would handle for various degrees of cooling. In addition to this the volumetric efficiency of the pump changes with temperature, but not in direct proportion to the volume changes. The vacuum, stroke, valve design, clearance, and leakage also influence the volumetric efficiency of a vacuum pump. Figure 4 shows the comparative volumetric efficiencies of a vacuum pump at various steam-air mixture temperatures, as found by actual tests. This curve is drawn up with air as a standard—that is, the volumetric efficiency is indicated in Figure 4 as a percentage of the volumetric efficiency of the same pump when handling dry air alone. It is evident from Figure 4 that up to about 68 degrees F. the volumetric efficiency of a pump handling steam-air mixture is the same as if the pump handled dry air alone. The volumetric efficiency of a certain vacuum pump handling air alone at 2 inches of mercury absolute pressure is about 56 per cent. This is the 100 per cent figure indicated in Figure 4. It is evident that for a cooling of 6 degrees (from 96 degrees to 90 degrees, as assumed) the comparative efficiency would be raised from 51 to 77 per cent of the dry-air value. The actual

FIGURE 1.—AUXILIARY SUPERCOOLING IN SURFACE CONDENSERS. —FIGURE 2.

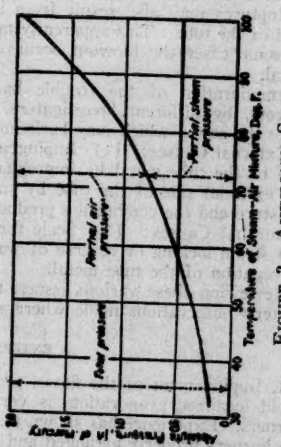


FIGURE 3.—AUXILIARY SUPERCOOLING IN SURFACE CONDENSERS. — FIGURE 4.

volumetric efficiency of the given pump under these conditions would be 0.51 × 0.56 and 0.77 × 0.56, or 28.5 and 43 per cent, respectively, a net gain of 14.5 points. In actual practice the yearly average of the cooling-water temperature is around 70 degrees F., giving mixture temperatures between 80 degrees and 85 degrees F. Under these conditions the average comparative volumetric efficiency of the vacuum pump, taken from Figure 4, would be between 94.5 and 88.5 per cent of the dry-air value, so that the actual volumetric efficiency of the given pump under these conditions would be 0.945 × 0.56 and 0.885 × 0.56, or 52.8 and 49.5 per cent, respectively.

It is therefore clear that the cooling of the residual steam-air mixture leaving a condenser before it reaches the vacuum pump is a powerful means of either increasing the capacity of the pump of an existing installation or reducing the size of pump required for a new job.—"Mechanical World," March 4, 1927.

WHY BOILER TUBES BLOW OUT.

By J. M. Brennan.

In modern power-plant practice the operation of water-tube boilers at high ratings has, under certain conditions, caused much trouble from blowouts. These ordinarily occur in the first row of tubes, where the water-heating surface is exposed to the highest temperature and to flame impingement. The greatest rate of steam generation takes place in these tubes, so that any cause tending to restrict the transfer of the furnace heat from metal to water, is likely to result in bulging and finally in a rupture.

Ruptures may also result from internal local corrosion, which thins the wall of the tube. This apparently takes place whether or not scale is present. In some cases the blowout occurs without any undue heating of the tube metal.

Amelioration of the trouble has been attempted, sometimes with fair success, by different investigators. They have advanced various theories as to the cause, which may be itemized as follows:

External Causes: (1) Impingement of furnace flame; (2) slow distillation of the tube metal by impregnation with the fuel oil; (3) corrosion of the external side of the tube by sulphuric acid, formed by the combination moisture and the combustion products of the sulphur in the oil used.

Internal Causes: (1) Scale-forming feed water; (2) grease on tubes; (3) steam locking or reverse of water flow; (4) corrosive feed water; (5) segregation of the tube metal.

Reviewing these various issues, the following notations are based on the writer's observations made where such conditions existed.

EXTERNAL CAUSES.

1. Impingement of the flame, producing a dry tube wall because of a too rapid localized evaporation, is very unusual and occurs only with fuel-oil burners. Experience has shown that when the furnaces are sufficiently deep, the burners properly arranged and adjusted and the proper care given to the tips, this action does not cause any trouble.

2. Slow distillation of the metal, or an oxidizing destruction by impregnation with the carbon residues of fuel oil depositing on the tube, is highly improbable. The high temperature in oil-burning furnaces prevents any carbon deposit on the fire side of the lower tubes. If any coke deposit

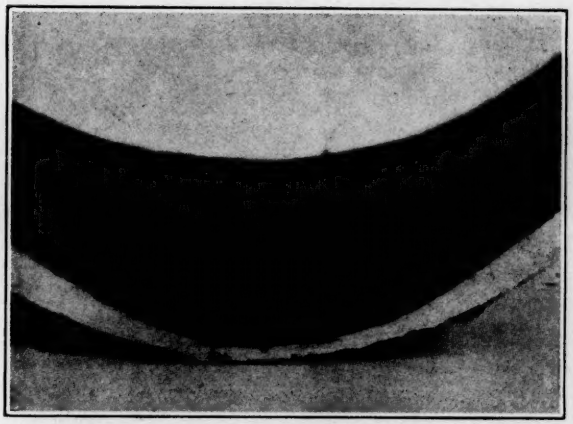


FIGURE 1.—THIS TUBE, OPERATED WITHOUT SCALE, SHOWS WASTING AWAY OF METAL ON INSIDE.

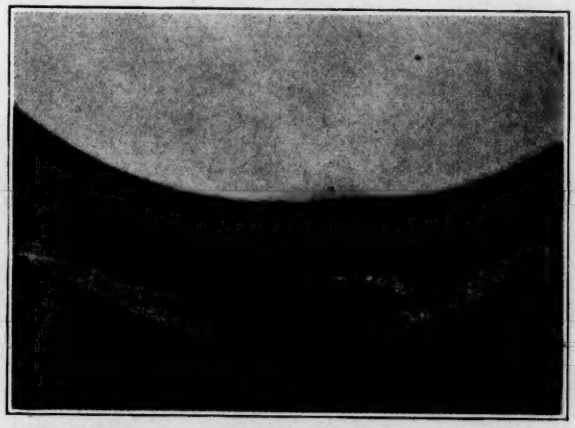


FIGURE 2.—AN EXTREME EXAMPLE OF SEGREGATION. HEAT RESISTING MATERIAL IN POCKET RESULTED IN BURNING OF METAL. THIS POCKET HAPPENED TO BE DUE TO A POOR LAP WELD.

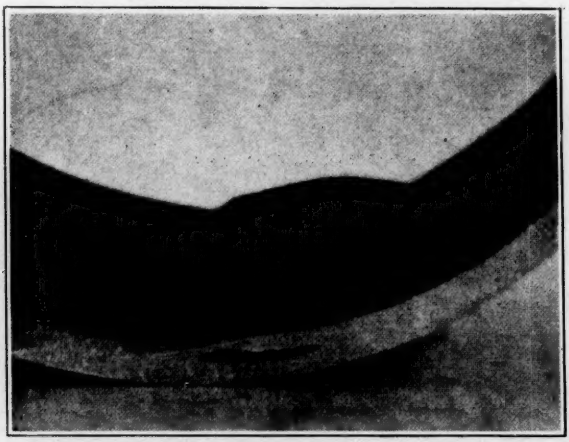


FIGURE 3.—OTHER END OF SPECIMEN PICTURED IN FIGURE 2, SHOWING SOME SEGREGATIONS HERMETICALLY SEALED BY THE TUBE METAL.

should occur on the upper side, it would have little effect. In Scotch-marine boilers where the flame strikes directly against the 5/8-inch metal of the Morrison tubes, no trouble occurs when the water side is clean.

It is possible, however, for this phenomenon to occur if heat flow from the tube to the water is so restricted that the metal reaches a high temperature. This can be caused by grease, corrosive layers, or scale of sufficient denseness, even though extremely thin. Under any one of these conditions damage to the outside of the tube occurs, forming a metallic scale similar to a crucible blister. This type of trouble produces very rapid destruction when once started.

3. Corrosion of the tube on the outside by sulphuric acid, formed from the gaseous combustion products, is practically impossible in the first rows of tubes, where the intensity of the furnace heat should prevent condensation of the acid. In the later passes of the boiler, due to leaky tubes or to blower steam mixing with the soot, corrosive action on the outside surface of the tubes will occur when boilers are not to be operated for a period, or when they are used on stand-by service, particularly in damp weather, the fire side of the tube should be cleaned. The reason is that if soot and ashes are allowed to cool with the tubes, they may form a scale which cannot be removed without wire brushes and metal scrapers. The fluffy deposit that clings to the tubes in oil-burning furnaces also carries sulphur. When a boiler is shut down, any leaks will dampen the solid deposits, which will then become corrosive.

INTERNAL CAUSES.

1 and 2. Scale or grease deposit in the tube is a prolific cause of tube failures because any condition that will permit the tube to reach a high temperature will cause bulging and a blowout. Transient scale, that is, a scale that will redissolve in the boiler when the temperature is lowered (as for instance when a boiler is standing by), is very elusive. If the boiler is opened after it has cooled down, no scale can be seen. Floating fragments, even though of eggshell thickness, can be redeposited in several successive layers, causing a local hot spot. At this point overheating will take place, leading eventually to bursting of the tubes.

3. The possibility of local overheating caused by the formation of steam bubbles on the under side of the tube is remote. The greater the rating of the boiler, the greater is the velocity of the water through the tubes. Since the generation of steam is greatest in the first row, the velocity is highest there. It is evident, therefore, that steam bubbles should be more quickly swept away. Moreover, these bubbles have an additional tendency to rise from the under side, due to the joint action of gravitation and currents in the water.

The reversal of flow in the boiler tubes and locking of a pocket of steam, which might leave the surface intermittently dry, is a somewhat obscure phenomenon. It is a question whether any such action takes place.

It is claimed that, with Stirling-type boilers, there is sometimes a spasmodic ejection of the steam, so that the tubes are not continuously wet. If such a condition is suspected, the concentration of flame at the lower end of the front tubes should be deflected by a new design of wall.

4. Corrosive feed water is a potential factor in causing tube blowouts. The deposit, formed at the expense of the tube metal, causes gradual thin-

ning. Corrosion products accelerate oxidizing. Being much poorer conductors of heat than the tube metal, they give rise to hot spots even when the tube is free from scale deposits.*

5. While those investigating tubes have given much thought to the treatment of the water before entering the boiler and within the boiler, little study has been made of the structure of the tube metal. Unfortunately, until very recently metals that are known to resist corrosion have not been obtainable in boiler tubes, owing to the greatly increased cost and difficulty of manufacture.

TUBE METAL SHOULD BE HOMOGENEOUS.

Recent investigators have found that segregations of the interior wall of tubes are a direct cause of internal local corrosion and tube blowouts. It has been found that sulphur prints taken from tubes selected at random showed the interior badly segregated. The belief is that these defects are due to pockets in the original ingots from which the steel was rolled. Rapid failure of tubes by corrosive feed water and the electrolytic action of metallic scales, segregates and blow holes, should be thoroughly investigated where blowouts occur in clean tubes.

It seems that the usual water tests for alkalinity and acidity are not quite adequate for the definite control of corrosion. Titration does not give any measure of the actual corrosiveness. On the other hand, the hydrogen-ion concentration† can be exactly regulated to the point where the smallest amount of corrosion takes place, with the addition of the least amount of alkali.

One large plant with the Stirling-type boiler used both steam and mechanical atomization for the fuel-oil burners. With the steam-atomizing type of boilers, failures increased at high ratings. The cause in every case was dirty tubes. Similar boilers at similar ratings showed more tube blowouts with mechanical atomization.

WATER TREATMENT OFTEN SOLVES DIFFICULTY.

After a thorough study of a record kept for a long period, it was found that when the front row of tubes was renewed no blowouts occurred until the new tubes had been in long enough to acquire a coating of scale. After each cleaning the blowouts diminished for a short time and then gradually increased up to the time of the next cleaning. Chemical treatment was then tried until finally a condition was brought about which left the tubes scale free. Thereupon the trouble ceased.

In a plant in which there were horizontal water-tube boilers, equipped with mechanical atomization, rapid burn-outs were laid to flame impingement and operation at too high ratings. As these boilers had been changed from steam atomization, where the trouble even at high ratings was slight, the new burner arrangement was suspected. But when the old scale was removed and chemical treatment was applied to keep the tubes clean it was found (even when operating at a continuous high rating) that the tubes ceased to bulge and blow out.

In a small industrial plant, severe surface corrosion of the boilers existed. No scale or local thinning could be found. An extensive investigation showed

* In some cases the velocity of water through the tubes is sufficient to sweep the corrosion residue away, giving corrosive feed water a better chance to attack the metal and hasten the destruction. The inside surface of such a tube will be found to be perfectly smooth and clean and thinned down at the blowout location.

† Stated simply, hydrogen-ion concentration is a definite measure of the actual "acidity" of a solution, regardless of what acids or alkalis it may contain. It may easily be determined by apparatus already on the market. Little chemical training or theoretical knowledge is necessary in order to use this control in practice.

improper operation of the open feed heater and proved that corrosive substances from process work were finding their way into the returns and back into the water. The various disturbing conditions were gradually alleviated and a chemical treatment applied to the boiler water to assist in checking the corrosion. As a result the difficulty ceased.

In some other boiler plants where blowouts occurred, an examination of boilers and sections of tubes cut open showed the surfaces to be clean. In one such case the cause of the tube blowouts was found to be an internal corrosion of the tube.

In the study of tube burning, as has been before mentioned, oil-fired furnaces were found to give the most trouble when mechanical atomization was used. Extended researches showed that the heat in the lower part of the furnace was more concentrated with mechanical atomization.

In studying the cases of internal corrosion, it is difficult to arrive at a conclusion that applies as a general rule, the cause usually being some local condition. If the types of tubes are studied and a record kept of the service of different types, and if both the feed water and the boiler water are kept free of scale forming and corrosive tendencies, it is believed that boilers can be operated at high ratings without any tube-blow difficulties. A bad stock of tubes might bring about a condition that would be very difficult to cope with, and therefore, before making any radical change in chemical treatment, it would be well to examine the metal of the tubes for flaws or segregations.

It might be advisable to use corrosion-resisting metal in the tubes next the furnace. Otherwise the mill scale should be removed by pickling or sand-blasting to prevent pitting and corrosion.—"Power," February 15, 1927.

ROLLING TUBES IN BOILER PLATES.*

By P. H. Oppenheimer, Dr.Eng.

Design Engineer, American Brown Boveri Electric Corporation.

Tube fastenings in boiler plates must meet two requirements: First, they must make a tight seal between fire and water, and secondly, they must withstand certain stresses that occur in all boilers under operating conditions due to the unequal expansion of the tubes and tube sheets. In certain types of boilers, such as the vertical water-tube type, the stresses due to the weight of drums suspended from the tubes must be added.

In principle a tube expander consists of three parts: A conical pin, an odd number of rolls, and a socket guiding the rolls. The task of the tube expander is to press the rolls radially against the inner wall of the tube by advancing the conical pin and to roll the tube on its entire periphery as the pin is turned.

TEST ARRANGEMENTS.

Experiments were conducted with tube expanders driven by a slow-speed electrical boring machine. The power input to the motor of the boring machine was measured by a recording wattmeter. Figure 1 represents a diagram of the input to the motor obtained in this manner.

This diagram shows that the process of rolling is composed of three operations:

a. Expanding the tube until it touches the wall of the hole in which it is being fastened.

* Abstract of thesis of the same title.

b. Rolling the tube and deforming the metal of the plate surrounding it.
c. Withdrawing the expander.

In expanding the tube (period *a*) the power input increases in proportion to the number of turns made by the tube expander. In Figure 1, it will be noted that as soon as the tube touches the surrounding metal, the power required increases rapidly, fluctuating widely to a maximum reached at the moment the motor is switched off. In period (*b*) the so-called rolling of the tube takes place.

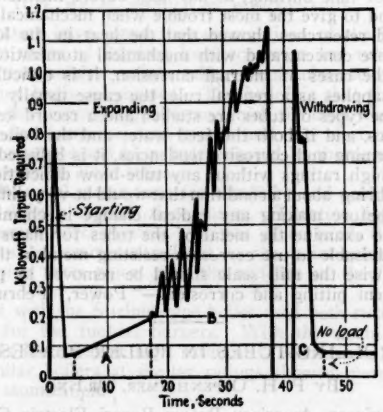


FIGURE 1.—POWER FOR ROLLING TUBE AND WITHDRAWING EXPANDER.

Tube rolling is a complicated case of cold rolling. Whereas, in rolling plates the plate is between two rolls, the distance between which may be considered as constant during each period of rolling, the tube to be rolled is between one roll of the tube expander on the inside and the material of the tube plate on the outside, the latter being deformed by the rolling. The layers of metal immediately surrounding the hole are deformed permanently. The more remote layers support the inner layers and undergo elastic deformations.

In the operation of the expander the pressure of each roll is transferred to the strip of tube metal in momentary contact with it. Most of the particles of metal flow in the direction of the tube axis, the direction offering the least resistance, causing the tube to elongate.

On both sides the tube is restrained from elongation by friction; on the inside, friction between the polished rolls of the expanding tool and the inner surface of the tube; on the outside, between the outer surface of the tube and the material of the plate. As the friction between rolls and tube is less than between plate and tube, the tendency to flow is greater on the inner surface of the tube than on the outer. The result is that the tube is more elongated on the inner side. By friction the tube carries along material on the brim of the hole in the plate in an axial direction. The value of this friction depends on the roughness of the tube and the hole.

Radial stresses arise in the tube and in the plate material surrounding the tube during the period of rolling, (*b*). The radial stresses in the tube will be small if the end of the tube is well annealed and the clearance of the tube in the hole was small before expanding.

The harder the original material of the tube and the more it is hardened by expanding on account of greater clearance, the higher the internal stresses of tube and plate will be. When the tube expander is withdrawn, the pressure of the rolls disappears, the internal stresses are released, hence the tube and the surrounding material of the plate spring back toward the center until equilibrium is reached; that is, until the radial forces normal to the exterior surface of the tube become equal to the opposite forces normal to the surface of the hole in the plate. Thus by rolling the plate shrinks upon the tube. In contradistinction to the heat-shrunk method of fastening, in this rolled-shrunk method the hole diameter is equal to the exterior tube diameter not only after having shrunk, but also immediately before shrinking. Thus the elastic shortening of the hole periphery is equal to the elastic shortening of the exterior tube periphery.

ANALYSIS OF THE HOLDING FORCE.

The holding force of a rolled tube against being drawn from the plate consists of the following resisting forces:

1. The friction between tube and plate due to the shrink fit.
2. In rolling a tube the part or end projecting from both sides of the plate is deformed more than the part of the tube within the hole. In loosening a tube projecting through the plate, the projecting end of the tube must be drawn through the hole of smaller diameter. This resistance may be increased by conical expanding, beading or rolling out by special methods.
3. The holding force is increased if it is necessary to shear parts of the tube or tube sheet to loosen the fastening. Such shearing occurs if a tube projecting through the plate is rolled very strongly or if the tube is rolled into circular grooves or with a turn of wire placed between the tube and tube sheet.

RESULTS OF TESTS.*

The material used for tests was new seamless boiler tubes and boiler plates, both of mild steel. Seven series of tests were made consisting of about 120 single tests. For each series of tests the pieces of tube were cut from the same tube if possible, while the flanges were cut by torch from the same steel plate.

The results of the tests of the plate material were:

Yield point	= 29,400 pounds per square inch
Ultimate tensile strength	= 47,400 pounds per square inch
Elongation at rupture	= 28.1 per cent
Reduction of area	= 71.8 per cent
Brinell hardness	= 92 to 112
Carbon content	= 0.043 per cent

The results of tests on the tube material as delivered:

Ultimate tensile strength	= 59,000 pounds per square inch
Elongation at rupture	= 23.8 per cent

* The tests were made by the author in the boiler plant of A. Rodberg A-G, and in the material testing laboratory of the Technical University, both in Darmstadt, Germany.

For the rolling tests pieces about 8 inches long were cut from stock tubes by a cold saw. The pieces were welded shut on one end over a mandrel turned to a hemisphere on the end. They were then heated in a charcoal fire and packed in ashes. After cooling, the scale was removed from the surface of the tube by sandblast or file.

The iron plates or flanges in which the tubes were rolled were 8 inches in diameter and were cut from the full plate by torch. Ordinarily, the boiler tubes used were of 3¼ inches external diameter and about ⅛ inch thickness (10 gauge).

After being rolled, the joints were tested for tightness in an apparatus able to apply hydraulic pressure of about 3000 pounds per square inch. During the hydraulic test the tubes were hammered. After that the holding force of each tube was determined in an Amsler-Laffon compression testing machine. The tubes were pressed out of the flange by means of the mandrel over which they were welded shut.

To determine the pure holding force of shrinking, the tubes were rolled while not extending beyond the plate. It was found by tests and also by mathematical analysis that the holding force of a shrunk fastening, the tube being simply rolled and not extending beyond the plate, depends on the following factors:

1. It is directly proportional to the coefficient of the slipping friction between tube and hole wall; that is, the holding force increases as the surface of tube and hole wall are rougher.

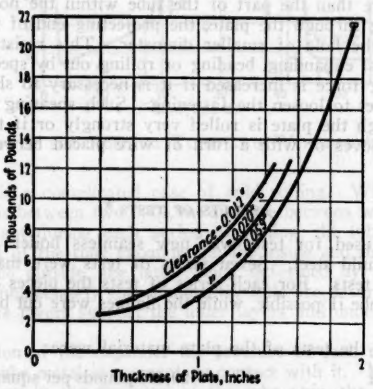


FIGURE 2.—HOLDING FORCE DUE TO SHRINK FIT ALONE IN RELATION TO PLATE THICKNESS.

2. It is proportional to the thickness of the plate. The curve of holding force increases less rapidly in thinner plates and faster in thicker plates, as shown in Figure 2, for different clearances between tube and hole wall before rolling.

3. It is a function of the amount the hole springs back after rolling, diminished by the amount the tube springs back. The elastic shortening of the tube after rolling is a direct function of the clearance between tube and hole diameters before rolling. See Figure 3.

4. It depends on the ratio of external radius to thickness of wall of tube after being rolled. As long as the radius is changed in the same ratio as the thickness, the shrinking resistance is not altered. Of tubes of an equal diameter, those with thicker walls have a greater holding force. Of tubes of the same wall thickness, those of the larger diameter have a smaller holding force, but a tube of larger diameter must be rolled by higher pressure than a smaller one to be tight.

5. It depends on the ratio of pitch to diameter of holes after rolling. The influence of this factor is smaller than the diameter-thickness ratio and of practical importance only if the pitch is very small.

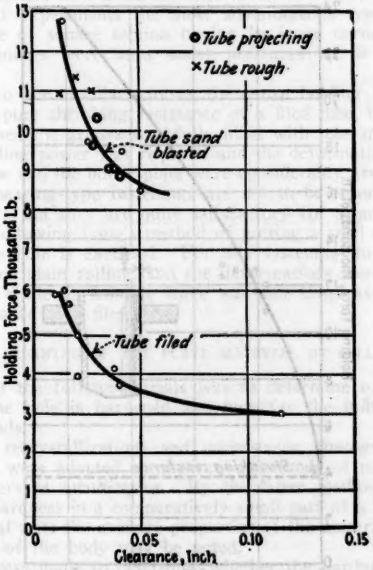


FIGURE 3.—EFFECT OF CLEARANCE ON HOLDING FORCE DUE TO SHRINK FIT ALONE.

The lower curve is for tubes with the surface filed smooth, the upper for sandblasted tubes. It is evident from both curves that the holding force increases considerably with decreasing initial clearance.

6. It is a function of the material of tube and plate and has its highest value if tubes and plates are of steel, and a smaller value if tubes are of steel and plates are of copper.

The power used for rolling has no influence on the holding force and the tightness of the shrunk fastening, after the power exceeds a certain minimum value depending on the diameter and the thickness of tube.

HOLDING FORCE OF A PROJECTING TUBE.

In being rolled by a cylindrical expander with the tube end projecting through the plate, the tube is pinched over the diameter of the hole on both sides of the plate in a radial direction. By the resistance of these parts of tube against being drawn or pushed into the smaller diameter of the hole, the entire holding force is increased considerably.

In this case the shaping of the edges of the holes is of great importance. If the edges are sharp, the holding force does not increase as long as the tubes are rolled by normal power. Figure 4 shows the holding force in relation to the radial projection of the tube end for filed tubes.

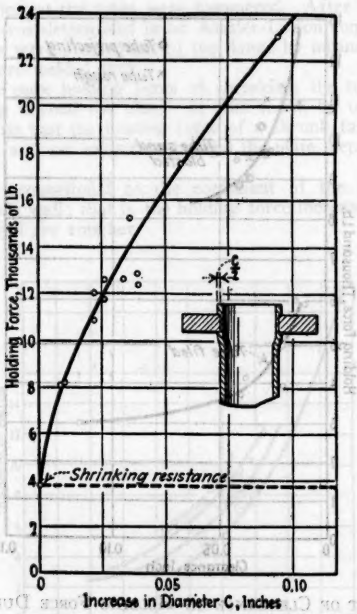


FIGURE 4.—EFFECT OF PROJECTING TUBE END ON HOLDING FORCE.

The average clearance is about 0.045 inch in diameter. The curve cuts the axis at 3750 pounds, the holding force due to the shrink fit for this clearance.

INCREASING THE HOLDING FORCE.

The resistance of the projecting tube end can be increased by different methods, such as conical expanding, rolling by means of special expanders, and beading. For this it is very important that the material of the tube be as soft as possible and the ends of the tubes to be expanded be annealed most carefully. Although the material of the tube end is hardened in rolling

by cold stretching, it must still be soft enough to bear the second process of expanding without splitting. The greater the angle at which the projecting end of the tube is expanded, the greater is the holding force. The maximum is reached in beading the tubes at an angle of 90 degrees.

When the projecting tube end is pinched over extremely, by using special expanders or by rolling with cylindrical rolls under very high pressure, the resistance of the tube against being drawn into the hole increases considerably, and the shoulder of the tube is sheared.

The most frequently used rolled fastening, based on shearing of the tube, is made by rolling into circular grooves. In this type of fastening one or several circular grooves are turned in the surface of the wall of the hole in the tube plate, and into these the tube metal is forced by rolling. By a great number of experiments the most advantageous type of groove was found to be one of square section having its inner corners rounded. All shear-type fastenings were tight under pressures up to 3000 pounds per square inch.

By rolling into one circular groove, the entire holding force is increased to 12 times the pure shrinking resistance of a filed tube, but this value was reached only when the groove was fully filled with tube metal. To do this, considerable rolling power was required and the deformations and the hardening of the tube and the boiler plate were considerably greater than in plain rolling. The shearing-type fastenings are not to be recommended for tube walls and boilers, but they are quite satisfactory for attaching pipe flanges.

A fastening following Lang's method of putting a steel wire between tube and wall of the hole is excepted. For this fastening no more power was required than for plain rolling, and the deformations were no greater than in plain rolling, while the holding force was nine times as much as the pure shrinking resistance of a filed tube.

HARDENING OF THE PLATE MATERIAL BY ROLLING.

The object of the following trials was to determine to what degree the plate around the hole is hardened and how far the influence of the cold stretching extends.

Strain lines, recrystallizations and microscopic photographs of the material structure were adopted as qualitative methods of test and the Brinell hardness numbers as quantitative. By the latter method it is possible to determine the hardness in a comparatively small part of a body, while by the other mechanical tests the average properties of the material over a more or less large part of the body may be noted.

First, tests were made to determine whether the hardness numbers found by the Brinell ball pressure test give a real measure for pronouncing judgment of the hardening of the material by cold working.

A piece of the boiler plate used for the trials was reserved for the tests of the material, out of which three strips along the direction of rolling were cut. Two test pieces were machined to bars of round section and one to a bar of rectangular section. The elongations of the first round bar were measured by the usual method, those of the second round bar exactly by Marten's mirror extensometer. The third bar of rectangular section, having on both sides the black mill scale, was given a certain load in the machine for testing tensile strength, taken out and given two ball pressure tests, then put in the tensile machine again, given a higher load, taken out, given two ball pressure tests, etc., until the bar failed. Also small cylinders were turned from the boiler plate and each compressed by a different load. The hardness in the center of the end surfaces of each cylinder was determined in the Brinell hardness testing machine.

The results of these tests are given in Figure 5. For comparison the dotted line found by A. Pomp is added, showing the increase in hardness of plates effected by cold rolling. From Figure 5 it is evident that hardening does not take place before the yield point is exceeded. In the tensile tests, the pressure tests and the rolling tests the hardness is directly proportional to the loads divided by the momentary section (in the tensile test only until the area of the bar is diminished rapidly). The hardness increases faster by tension than by pressure. From this proportion and from the fact that the curve of hardness coincides with Ludwik's flow curve, it was concluded that the hardening of the same material by cold working is determined by the Brinell hardness numbers.

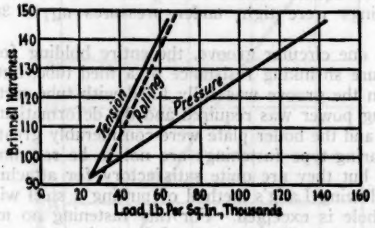


FIGURE 5.—BRINELL HARDNESS AS AN INDICATION OF STRESS SUSTAINED.

To investigate the hardening of the plate material by rolling, the tube holes were turned out eccentrically and the flanges were divided into four parts by radial cuts as indicated in Figure 6, so that the single pieces of plate could be tested conveniently by the ball hardness test. The distance of each point of measurement from the original hole wall was found as shown in Figure 6.

The curves of hardness, one of which is given in Figure 7, are similar. Reaching the greatest hardness at a small distance from the wall of the hole, they decrease rapidly to the value of the original hardness of the material. By this method the material of 15 flanges was tested in the vicinity of the hole, and it was determined that the hardening grows approximately proportional to the power required by rolling.

INSPECTION OF TWO TUBE SHEETS IN EACH OF WHICH FIVE TUBES HAD BEEN ROLLED.

How several tubes rolled in the same plate influence each other, is of special interest to the boilermaker. Therefore, two series of tests, A and B, were made. In each tube sheet of 16-inch diameter five 3¼-inch tubes were rolled, one in the center and four around it. Their location is evident from Figure 8.

The average breadth of the bridge of plate between the central tube and the exterior tubes was 15/16 inch. The tests differ only in the required rolling power, the average being in series A, 1.27 horsepower, in series B, 2.28 horsepower. To reveal plainly the strain figures, the front side of each tube plate was polished.

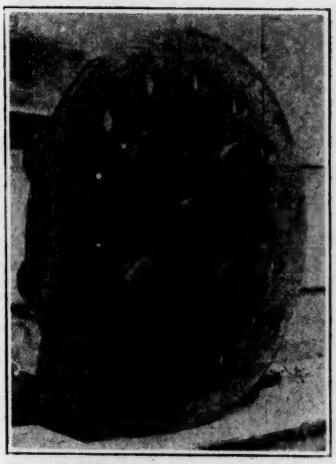


FIGURE 8.—TUBE SHEET WITH TUBES ROLLED IN.

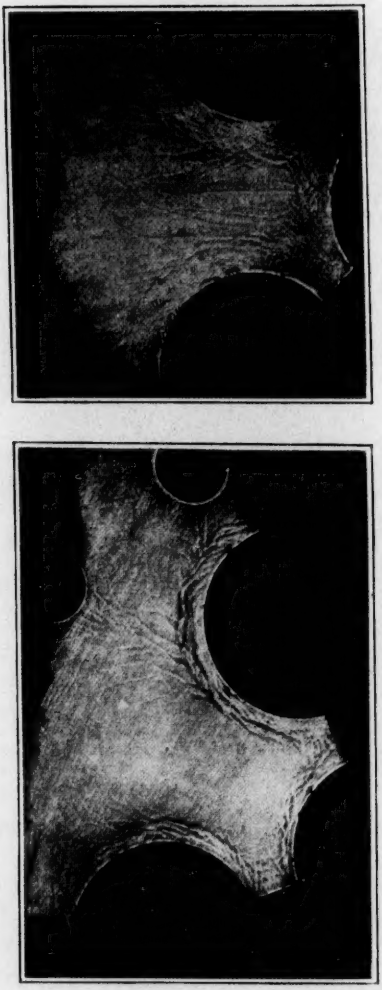


FIGURE 9.—STRAIN LINES ON FACE OF TUBE SHEETS.
Upper, strongly rolled series B; lower, normally rolled, series A.

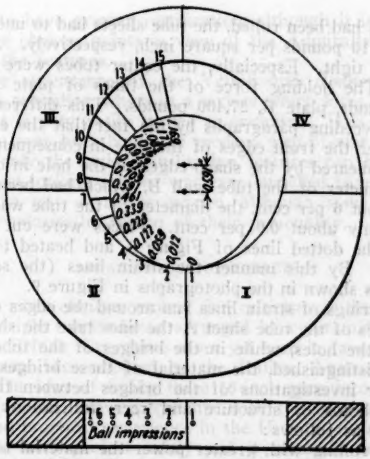


FIGURE 6.—FLANGE PREPARED FOR BALL HARDNESS TESTS.

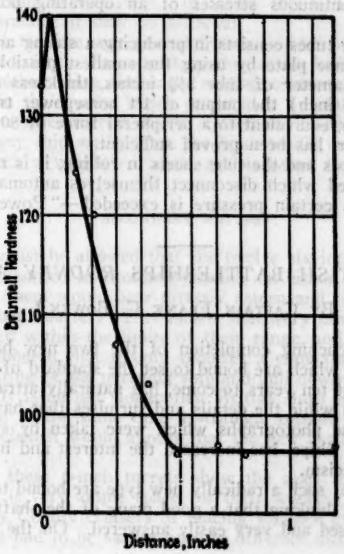


FIGURE 7.—BRINELL HARDNESS AT VARIOUS DISTANCES FROM TUBE WALL.

After the tubes had been rolled, the tube sheets had to undergo a hydraulic test at 625 and 710 pounds per square inch, respectively. All fastenings in each group were tight. Especially, the center tubes were completely tight in both cases. The holding force of the tubes of plate A was found to average 9400 pounds, plate B, 27,400 pounds. This difference has been explained in the preceding paragraphs by the fact that the ends of the tubes being pinched over the front edges of the hole in consequence of the strong rolling, must be sheared by the sharp edges of the hole in order to be withdrawn. The diameter of the tube wall B, which had been rolled strongly, had increased about 6 per cent, the diameter of the tube wall A, rolled with normal power, only about 0.9 per cent. Pieces were cut from each tube wall, following the dotted lines of Figure 8, and heated to 570 degrees F. in a bath of oil. By this manner the strain lines (the so-called Lueders lines) appeared as shown in the photographs in Figure 9.

It looks as if strings of strain lines run around the edges of the holes. In the narrow bridges of the tube sheet A the lines take the shortest way from edge to edge of the holes, while in the bridges of the tube sheet B strain lines cannot be distinguished, the material of these bridges being hardened entirely. Further investigations of the bridges between the holes by ball hardness tests, photos of structure and recrystallizations gave the same results, which are as follows:

As a result of rolling with greater power the material of the bridge between the tube holes becomes considerably different from the original material. It becomes more brittle over the whole section; that is, the hardness, yield point and tensile strength have increased, while the elongation has decreased. In other words, the material of the plate has become less qualified for bearing the continuous stresses of an operating boiler than it was originally.

The art of rolling tubes consists in producing a strong and tight fastening between tube and tube plate by using the smallest possible power. In the tests mentioned (diameter of tube 3¼ inches, thickness of tube ⅛ inch, thickness of plate 1 inch) the output of 1.1 horsepower transmitted on the periphery of the pin, equivalent to a peripheral force of 800 pounds on each roll of the expander, has been proved sufficient.

To protect the tubes and the tube sheets in rolling, it is recommended that expanders be adopted which disconnect themselves automatically or give a signal as soon as a certain pressure is exceeded.—"Power," February 22, 1927.

THE NEW BRITISH BATTLESHIPS *RODNEY* AND *NELSON*.

By Captain Frank C. Bowen.*

The rapidly approaching completion of the two new battleships *Nelson* and *Rodney*, vessels which are bound to set the standard of capital ship construction for at least ten years to come, has naturally attracted a good deal of attention to them while the details and surmises that have been published in the press and the photographs which were taken by special permission for Jane's Fighting Ships, has increased the interest and has brought about a good deal of criticism.

Of course, ships of such a radically new type are bound to invite criticism, but one cannot help thinking that a good many of the shafts that are aimed at them are ill-advised and very easily answered. On the other hand some

* London Correspondent to the "Marine Journal."

of them appear to have greater justification, although it is only possible to examine the more obvious points in view of the Admiralty's reticence as to the majority of the features.

Admittedly the appearance of the *Nelson* and *Rodney* is sufficient to arouse immediate prejudice, which in a service which loves its ships as much as the Navy does is very easily turned to criticism. It is doubtful whether uglier ships have ever been built, but modern men-of-war are not designed for beauty and only the strictly utilitarian features can be considered. A number of the best-looking men-of-war ever launched have turned out utter failures under the test of active service, although some of them always maintain their popularity by reason of their personality.

ARRANGEMENT OF TURRETS.

The first and most obvious ground for criticism is the grouping of the guns, the three triple turrets being arranged in a pyramid well forward of the bridge. The British Navy have never yet been in favor of triple turrets, although they have been adopted by the Italian, Austrian, American and other Navies. The idea of fitting three 16-inch guns into a single turret is certainly a daring one, but experiments with the monitor *Lord Clive* off Shoeburyness and practical experience in the U. S. S. *Colorado* suggest that it could be done with perfect safety and it certainly permits remarkable concentration of fire, which in modern battle tactics is half the struggle. In addition to the principle of triple turrets their arrangement has come in for a good deal of adverse comment, it being pointed out that although the whole battery of nine guns will bear on the broadside only six can be fired right ahead and none at all astern. In theory this is a very serious disadvantage but in practice it may be different.

Experience with the late war proved that no matter how keen the chase it was seldom right astern. For one thing the smoke had a very considerable influence, for another thing the early raiding cruisers made a practice of dropping mines in their wake which rendered it advisable to keep well out on the beam. The three turrets of the *Nelson* and *Rodney* have such a large arc of fire that any ship well out on the beam would come under their concentrated fire, such a weight of metal as has never before been attempted at sea.

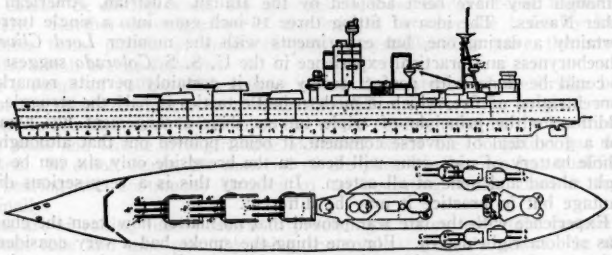
SECONDARY BATTERY.

In addition it must be allowed that the twelve six-inch guns are arranged in twin turrets mounted in similar pyramids on either side of the ship aft, so that eight of them would bear directly astern and over a very considerable field on either quarter. This means that very heavy and rapid salvos could be delivered within the limits of their range, and in a pursuit it must always be remembered that it only requires a comparatively small shell to open the bow plating of the pursuing ship and pull down her speed by several knots at once. It is an old slogan that British ships have never been built to run away, but at the same time a strategic retreat is often wise and in spite of the criticism it would appear that the new battleships are quite able to effect it.

The ports of these 6-inch turrets show the guns to be wide apart and apparently there is very ample space for their working, the unlucky experiment in the early County cruisers of placing six-inch guns so close together that the breeches had to be swung open in different directions having taught its lesson.

FACILE FIRE CONTROL.

This concentration of the main armament of the ship over a comparatively short section of her length will greatly facilitate fire control, a very important matter in modern warfare. It also simplifies the magazine arrangements and makes them considerably safer and easier to protect without a huge expenditure of weight that might otherwise be given to armament and speed. Of course, it lays itself open to the suggestion that we are putting too many eggs into one basket, a favorite catch-word ever since any attempt has been made to group the armament of a man-of-war, and many would have us believe that a single lucky hit will put the entire main armament out of action. One cannot say that such a thing is impossible—nothing is impossible in modern naval warfare—but one cannot help thinking that it would have to be an extraordinary salvo placed in extraordinary fashion to effect such a result. The *Nelson's* guns are as nearly invulnerable as human ingenuity could make them and the possibility of such a catastrophe is most remote.



Profile and Deck Plan of the "Rodney" and "Nelson"

Also there is the advantage that the enormous weight of the guns, their mountings, turrets, ammunition supply and everything that goes to make a modern piece of naval artillery are placed in the broadest section of the ship, and this avoids the tendency to pitching which mars the seaworthiness of so many heavily armed men-of-war.

TORPEDO DEFENSE.

On the question of anti-torpedo armament the critics certainly seem to have something in their favor. Big ships are not often directly ahead, but torpedo craft can attack from every point of the compass and the position of the battery of six-inch guns would appear to make it very difficult to concentrate a straight-ahead fire at short range. Certainly the searchlight equipment is so placed and so powerful that the attackers would almost inevitably be discovered long before they got into that range, but the possibility must not be neglected. It may prove, however, that some of the 4-inch anti-aircraft guns, whose position has been keep a close secret by the Admiralty, will be available to help resist destroyer attack.

ANTI-AIRCRAFT BATTERIES.

The anti-aircraft battery is another subject on which there is likely to be a good deal of discussion when full details are known. The Admiralty has published the fact that it is to consist of ten 4-inch guns but there are no

details as to how they are to be placed, although there are many rumors afloat of an entirely novel and very interesting plan. The question is whether the caliber is sufficient. The only two classes of ships which can in any way be described as being contemporary to the *Nelson* are the Japanese and American aircraft carriers, and of these the former mount twelve 4.7-inch high angle guns apiece and the latter twelve 5-inch. There is thus a very considerable difference in the weight of metal, and one of the most frequently made aspersions on British design is that we incline to protect our ships far too lightly against air attack, many sizeable ships being provided with 3-inch guns only.

Of the torpedo armament of the new ships it is impossible to speak, for it is not even known whether they carry any. Rumor has it, and it has attained very considerable credence in foreign naval circles, that they are to be built without any torpedo tubes at all and considering the very small use that the capital ships made of this weapon during the war it is not at all improbable that this will prove to be true. It is unwise to judge, however, until further details are published.

Externally the most striking feature of the new ships is undoubtedly the huge director tower which combines the function of the bridge and tripod mast and is believed to be far more efficient. It would seem to add a lot to the topweight of the ship and also to be terribly vulnerable to shell fire, while if it is possible to approach these ships at comparatively short range it will undoubtedly offer a big target. There is also a mystery tower aft, whose purpose has not yet been discovered, although it has been the subject of infinite discussion.

BELT AND SIDE ARMOR.

It is believed that the main protection consists of a 14-inch belt, arranged on the "all or nothing" principle which is so popular in the U. S. Navy and which is gaining favor all over the world. The arrangement of the main and secondary batteries assists this principle, while it is obvious from the photographs that the Admiralty have allowed to be taken that it will not extend very much above the water-line and that except for the bases of the 6-inch turrets there is no other side armor. If the ship is fitted with blisters they are internal, but it is known that her hull is very minutely sub-divided as a protection against under-water explosion and that thick and well-arranged decks have been fitted against aircraft bombs.

SPEED POSSIBILITIES.

Perhaps the greatest riddle of all, however, is the speed of these ships. Well-informed naval circles some time ago were suggesting that it would be between 22 and 23 knots only, while latterly there seems to have been a growing impression abroad that they will be 25-knot ships designed to operate alongside the *Queen Elizabeth* type, who certainly justified their speed during the war. As no other power can build capital ships until the end of the Washington Agreement it is probable that the Admiralty will publish fairly full details when they pass their trials, but until then either of these opinions has an equal right to be considered.

Taking them all together, and omitting reference to their truly hideous appearance, it would appear that the *Nelson* and *Rodney* are far and away the finest ships afloat and are well worthy of the reputation of Sir Eustace Tennyson D'Eyncourt, who designed them shortly before he resigned the post of Director of Naval Construction. They were designed in peculiar circumstances with the construction of capital ships restricted so drastically

by Washington, but taking these circumstances into consideration it would appear that they are capable of fulfilling their functions excellently—provided one does not accept the suggestions that a pursuing enemy will lie directly in the wake of the ship and that a single lucky shell can put out three of the heaviest turrets ever put afloat, turrets whose bases are placed quite a considerable distance apart owing to the size of the guns that they mount and the ammunition that they fire. Certainly one hears very much more criticism of the design in England than one does abroad, where they have apparently been accepted as the standard of the big ship for years to come.—"The Marine Journal," February 12, 1927.

THE MACHINING OF STAINLESS STEELS.*

By Walter Locke.

The casting, forging, and heat treatment of stainless and non-corrodible steels, after considerable research work, is now thoroughly understood and is quite a sound commercial proposition. It is only when they get into the hands of workmen for the first time that difficulties and troubles arise.

The reason why a skilled mechanic, who knows that a different form of tool and cutting speeds is required to machine cast iron and wrought iron, should insist on using the same form of tool and speeds to machine a mild steel and an alloy steel having a much greater difference in mechanical properties than there is between cast iron and wrought iron, has always been difficult for me to understand. The fact that it is steel is no sound reason for keeping to the same formation with regard to shape and speeds. At the same time, too much blame should not be attached to the mechanic, for the simple reason that information, which should be, and I venture to say always is, in the possession of those who purchase and the the designer who proposes to use this or any other class of new steel, is not passed on to those in control of production in the works, much less to the workman who has to produce the finished article. I am speaking from actual experience in several workshops.

The point is this—and it is a very important one—trust the man in the works and give him all the information you yourself possess; it will repay a hundredfold.

Our first production experience was to machine the surface of all ingots, either by planing or milling, as it was found that to produce a perfect steel, free from blemish when finished, all surface marks (even very small pinholes) must be removed.

The ingots are invariably square in form. Having suitable planing machines we set off by planing, and for some time we had difficulty in getting the output necessary to keep the production cost within reasonable limts. When we got our machine men educated to appreciate the difference in the nature of the steel, and experiment with the shape of tools and cutting speed, we made rapid progress.

The average time now taken to machine two ingots 12 inches square by 4 feet long with approximately 1½-inch radius on the four corners, removing not less than ¼ inch and up to ⅜ inch in depth of cut, is seven to eight hours, against 24 to 30 hours for two when we first started.

* From a paper read before the Glasgow and West of Scotland Association of Foremen Engineers and Draughtsmen. Mr. Locke is machine-shop manager of Thomas Firth and Sons Limited, Sheffield.

The alteration in the shape of the tool was surprisingly small—just a slightly increased backward slope—*i.e.*, the cutting-point behind the top of the tool to prevent digging-in and to break away the sand usually left round the feeder-head, a slight increase in the side-top rake, a slightly modified radius on the point, and away we went. No one was better pleased than the machineman when we got going; it meant less grinding (the thing that all good machinists hate as a loss of time) and a considerable increase in his wages as a result of an honest piecework price.

The shape of the tool used is shown in Figure 1. The cutting speeds require more change than the shape of the tool. We run on what is known as the standard F.G. Grade between 25 feet and 30 feet per minute on the

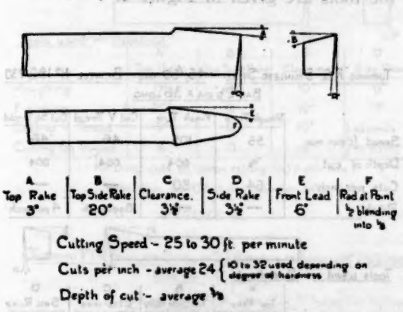


FIGURE 1.

hard grades to come down to 18 feet to 20 feet per minute. The traverse we vary from 32 per inch to 10 or 12 per inch—naturally the finer traverse on the hard material. If we got the ingots entirely free from sand on the feeder-heads, much higher speeds could be used.

We now come to the stage when most of those present first deal with these steels in the form of bars, forgings, or castings. Usually the first operation is turning. In our own works we have a number of lathes always on this work, machining it in every grade where machining is required. Take the most popular grade used amongst engineers—*i.e.*, "F.G." in the 45/50 tons tensile condition with a Brinell number of 190/230. Say we are turning a bar 2¼ inches diameter by 3 feet or 4 feet long, the finer feed being used on longer and more springy work, our roughing speed would be 55 feet to 60 feet per minute, traverse between 32/64 per inch, depth of cut ⅛ inch a side.

For finishing (note the difference from the usual practice in ordinary steels) a speed of 100 feet to 110 feet per minute is used, with a depth of cut of 0.004 inch a side, and a traverse of 80 per inch. It makes no appreciable difference whether a lubricant is used or not; we ourselves rarely use it. At this speed a beautiful clean surface is left, easily polished by emery or a lead lap.

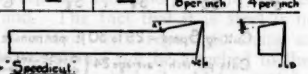
Threads cut in the lathe give equally good results. Cutting a vee thread 8 per inch we run at 45 feet to 50 feet per minute, depth of cut 0.004 inch, number of cuts to produce thread, 20, last two cuts to get smooth finish 100 feet per minute, letting the tool just skim the one side each time up. For cutting a square thread, 4 per inch, the same speed and depth of cut are used.

Of course there must be a sound reason for this relatively high speed and fine feed, especially on finishing. To me the real reason is the draggy nature of the steel when being machined, no matter what the operation is. When I have explained to a workman, who is machining this steel for the first time, that he will make most progress if he shapes his tool, uses feeds and speeds similar to what he would use for wood-turning, he appreciates the point that the material must be shaved off, not shorn off in the usual way. Needless to say, he is also told not to take this instruction literally, but as a guide on how to proceed with his operation.

The tools used are ordinary "Speedicut" high-speed steel, and are ground. The shapes of the tools are given in Figure 2.

Turning "F.G." Stainless Steel - 45/50 tons - Brinell No 190/230
Bar 2½ dia x 36" long.

	Rough Turn	Finish Turn	Cut V Thread	Cut Sq Thread
Speed ft per min.	55	105	46	46
Depth of cut	⅛	004"	004"	004"
Cuts per inch	64	80	—	—
Pitch	—	—	8 per inch	4 per inch



Tools used - "Speedicut"

	A Top Rake	B Top Side Rake	C Clearance	D Side Rake
Roughing	13°	13°	10°	8°
Finishing	20°	0	12°	7°
Cutting V Thread	8°	0	15°	depends on pitch
Cutting Sq Thread	8°	0	12°	

Finishing cuts on screw cutting 100 ft per min.
Work done on 10" centre toolroom lathe.

FIGURE 2.

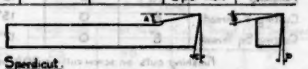
The same angles are used for all grades of stainless steels except in the case where the mechanical condition is over 55 tons, when slightly less rake is given to the roughing-tool, the angles being: top rake, 10 degrees; top side rake, 10 degrees; side clearance, 7 degrees; front clearance, 9 degrees.

When the steel is in the 50/55 tons condition with a Brinell number of 220/250, for roughing we run at 45/50 feet per minute, using the same traverse as before; for finishing, 90 feet per minute instead of 110 (Figure 3). In turning a bar 1⅝ inches in diameter by 63 inches long of "F.G." steel in the 55/60 tons condition with a Brinell number of 250/280, we use a tool of slightly less rake, as already explained. In roughing, the speed was reduced to 40/45 feet per minute, still using the same traverse; for finishing, 80 feet per minute with 0.004 inch depth of cut and 80 cuts per inch (Figure 4). For cutting either vee or square threads, 30 feet to 35 feet per minute, the finishing cut being taken at 80 feet per minute.

Now comes one of the peculiarities of stainless. In turning a bar 45 milimeters diameter of "F.M." (which is almost a stainless iron) we had to reduce the speed from that used on the softest grade of "F.G."—*i.e.*, from 55/60 feet to 45/50 feet per minute for roughing, and a slightly less depth of cut (3-32 inch instead of ⅛ inch a side), but used the same traverse. In finishing, 80/85 feet per minute instead of 100 to 110. Cutting vee and

Turning F.G. Stainless Steel – 50/55 tons – Brinell N°220/250.
Bar 1⅝ dia. x 42 long.

	Rough Turn	Finish Turn	Cut V Thread	Cut Sq Thread
Speed ft per min	45	90	38	39
Depth of cut.	⅛	·004	·004	·004
Cuts per inch	64	80	—	—
Pitch	—	—	8 per inch	4 per inch



Tools used – Speedcut.

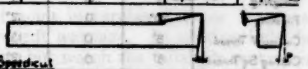
	A Top Rake	B Top Side Rake	C Clearance	D Side Rake
Roughing	15°	15°	10°	8°
Finishing	20°	0	12°	7°
Cutting V Thread	8°	0	15°	depends
Cutting Sq Thread	8°	0	12°	on pitch

Finishing cuts on screw cutting 90 ft per min.
Work done on 10" centre toolroom lathe

FIGURE 3.

Turning F.G. Stainless Steel – 55/60 tons – Brinell N°250/280
Bar 1⅝ x 63 long.

	Rough Turn	Finish Turn	Cut V Thread	Cut Sq Thread
Speed ft per min	42	80	34	34
Depth of cut	⅛	·004	·004	·004
Cuts per inch	64	80	—	—
Pitch	—	—	8 per inch	4 per inch



Tools used – Speedcut

	A Top Rake	B Top Side Rake	C Clearance	D Side Rake
Roughing	10°	10°	9°	7°
Finishing	20°	0	12°	7°
Cutting V Thread	8°	0	15°	depends
Cutting Sq Thread	8°	0	12°	on pitch

Finishing cuts on screw cutting 80 ft per min
Work done on 10" centre toolroom lathe

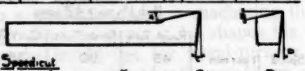
FIGURE 4.

square threads were done at 35/40 feet per minute, the finishing cuts being taken at 80 feet per minute (Figure 5).

On a bar of 1½-inch diameter S.A.P. steel, the roughing speed was 25/30 feet per minute, 1-16 inch a side depth of cut. Finishing was done at 50 feet per minute, 0.004 inch depth of cut, and 80 cuts per inch. Vee and square threads were cut at 28/30 feet per minute (Figure 6).

Turning FM Stainless Steel – 35/40 Tons – Brinell N° 200/220
Bar 1¾ dia x 42" long

	Rough Turn	Finish Turn	Cut V Thread	Cut Sq Thread
Speed ft per min	46	82	36	38
Depth of cut	1/32	004	004	004
Cuts per inch	64	80	—	—
Pitch	—	—	8 per inch	4 per inch



Tools used – Speedicut

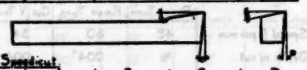
	A Top Rake	B Top Side Rake	C Clearance	D Side Rake
Roughing	13°	15°	10°	8°
Finishing	20°	0	12°	7°
Cutting V Thread	8°	0	15°	depends on pitch
Cutting Sq Thread	8°	0	12°	

Finishing cuts on screw cutting 80 ft per min
Work done on 10" centre toolroom lathe

FIGURE 5.

Turning 'SAP' Stainless Steel – 45/50 tons – Brinell N° 180/190
Bar 1¾ dia x 54" long

	Rough Turn	Finish Turn	Cut V Thread	Cut Sq Thread
Speed ft per min	26	48	30	28
Depth of cut	1/16"	004"	004"	004"
Cuts per inch	64	80	—	—
Pitch	—	—	8 per inch	4 per inch.



Tools used – Speedicut

	A Top Rake	B Top Side Rake	C Clearance	D Side Rake
Roughing	13°	15°	10°	8°
Finishing	20°	0	12°	7°
Cutting V Thread	8°	0	15°	depends on pitch
Cutting Sq Thread	8°	0	12°	

Work done on 10" centre toolroom lathe.

FIGURE 6.

On a bar of Staybrite steel 2 inches diameter, 44 inches long as rolled, the roughing speed was 23 feet per minute, depth of cut 3-32 inch, feed 1-64 inch. Finishing cut 45 feet per minute, depth of cut 0.004 inch, feed 80 per inch. Cutting vee and square threads, cutting speed 20 feet per minute, depth of cut 0.004 inch (Figure 7).

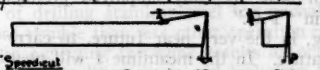
On bars 4 inches and 4½ inches diameter, heat-treated, the roughing speed was 45 feet per minute, feed 40 cuts per inch, depth of cut 3-16 inch. Unless this class of steel is heat-treated after forging or rolling, considerable variation may be expected in the speed at which it may be machined. The explanation is that cold working hardens this steel considerably more than in any other grade.

Although in the examples given we used approximately 64 cuts per inch for roughing, it does not mean that that feed should be rigidly adhered to—it happened to be the most suitable on the lathe used. The next feed of 30 cuts was not so satisfactory, but generally speaking it will be found the best results on light work can be obtained by using a feed of 40 to 60 cuts per inch, depending on the diameter and length of the bar or forging.

On forged rolls of 14 inches to 16 inches diameter, 8 feet to 10 feet long, we used tools of approximately the same angles. The depth of cut was ⅜ inch a side, feed 10 and 12 per inch, with very similar cutting speeds—*i.e.*, 55/60 feet per minute.

Turning FSB Stainless Steel — 50/55 tons — Brinell No 185/200
Bar 2" dia x 44" long

	Rough Turn	Finish Turn	Cut V Thread	Cut Sq Thread
Speed ft per min.	23	45	20	20.
Depth of cut	1/16"	.004"	.004"	.004"
Cuts per inch.	64	80	—	—
Pitch	—	—	8 thds per in.	4 thds per in



Tools used - "Speed-cut"

	A Top Rake	B Top Side Rake	C Clearance	D Side Rake
Roughing	13°	13°	10°	8°
Finishing	20°	0	12°	7°
Cutting V Thread	8°	0	15°	depends on pitch
Cutting Sq Thread	8°	0	12°	

Work done on 10" centre Lockroom lathe

FIGURE 7.

There is one important point in turning bars or forgings of any length: care must be taken to prevent warpage. In turning light bars use the finer feed and keep the speed up, but watch the tailstock center—instead of having to tighten it as usual you will have to slacken it considerably, owing to expansion. In the early stages of turning bars and forgings of any length, this point used to give us considerable trouble, but after the turners had realized this point, the trouble has been almost, if not entirely, eliminated.

If very fine limits are required it is essential that the material should be rough-machined to nearly finished dimensions and then laid down for a period, this period depending on the size, etc., of the forging or casting, the reason being to allow the mass to come to an even temperature.

Our firm, being essentially steelmakers and not general engineers, we had practically no use for a turret lathe, but to put us in the position of being able to demonstrate that stainless steels could be machined in this class of lathe as a commercial proposition, our directors decided to install one or more turret lathes.

The first test run under my observation was to turn the 1⅛-inch-diameter Staybrite bar down to 9-16 inch diameter, the speed used being 46 revolutions per minute with a feed of 125 cuts per inch. The surface was fair but pitted, probably due to scale getting between the roller steadies and the

bar. It was then screwed at the same speed, the pitch being 16 threads per inch, and the thread produced was very rough and draggy. They then ran a solid die up the thread; this gave a fairly smooth top, but the sides were still draggy.

For the next test I suggested we should try a much higher speed, and increased top-side-rake on the turning tool, and also increased top-rake on the dies. Taking the same depth of cut and the same speed—*i.e.*, from 1⅛ inch diameter to 9/16 inch diameter with a feed of 125 cuts per inch on the same bar—the speed was put up to 272 revolutions per minute, giving a surface speed of 80 feet on the 1⅛ inch diameter. The result on the turning was excellent, a good smooth surface being produced. The thread was cut at the same speed, when again a much improved result was obtained. Another sample from the same bar was turned at the same feed and speed, but the speed for screwing was put up to 416 revolutions per minute. This speed gave an excellent thread, but the speed was too great to give any life with the dies.

We have now got four turret lathes installed, but have found a production job for them which has given us practically no opportunity of carrying out further tests except on one machine. This machine, a 5½-inch ungeared capstan lathe, is fully employed producing two different types of screws made in "F.G."

I am hoping, in the very near future, to carry out a series of tests of a very varied nature. In the meantime I will mention only one which I have made, just to show the unexpected difficulties which crop up with this class of steel.

A large inquiry came in for Westinghouse brake piston-rods made in "F.G." stainless steel, to be finish-machined complete with cotter holes. I naturally said to myself, "This is an ideal job for the turret lathes," so before quoting I decided to run a test, when a new difficulty was discovered. These rods are 1½ inches diameter by 20 inches long, and I proposed making them from rolled bar. In turning, taking ⅛ inch a side depth of cut, with a feed of 200 cuts per inch, the surface left by the tool and roller steadies was disappointing. There was not enough pressure on the roller steadies to give the burnished finish I expected, and the new difficulty I found was that, after traveling 6 inches or 7 inches along the bar, the swarf was carried round, getting between the roller steadies and the bar, and causing the bar to dig into the tool. The tool was reground several times to different angles, but owing to the swarf coming off in long strips, and getting more or less crushed in the box-tool holder, the result was practically the same in each case, a dig-in of the bar on to the tool.

Another trouble was the large amount of very small pieces (almost like fine scale, but really very fine pieces coming from the swarf) being carried round, with the oil getting between the roller steadies and the bar, giving a pitting effect on the surface several thousandths deep.

A good finish was obtained by taking a light cut of from 1/32 inch to 1/16 inch a side at a cutting speed of 120 feet per minute and 200 cuts per inch. The cut being light, we did not get the good burnished finish that is obtained on mild steel. The surface, when examined with a good glass, was perfectly smooth and clean, the traverse of the tool being easily followed all the length machined, but after polishing with fine emery cloth all trace of the traverse was removed.

After seeing this test I concluded that further experiments were necessary before I could say turning long lengths in a turret lathe was a success, but I am certain it is only a question of further tests to get a satisfactory result.

From turning and screwing we will now turn our attention to some of the other machining operations. Take milling in its various forms:

1. *Slab Milling.*—Neither in the 45/50 nor 50/55 tons is there any difficulty in milling. Rigidity in holding is a very important factor, and a sharp cutter is essential to a good finish. As a commercial proposition we suggest a cutter speed of 55 feet per minute, depth of cut ⅛ inch or more if machine is sufficiently rigid and powerful, and a feed of 3 inches per minute. A slow-spiral cutter should not be used. A quick spiral has much superior cutting qualities. Lubricant used: 1 part soluble oil, 50 parts water.

2. *Milling Stainless Turbine Blading.*—In milling either the concave or convex face of the blade, the same remarks hold good as for slab milling except as regards depth of cut, which, owing to close rolling operations, is not likely to exceed 1/16 inch. This ensures a good finish on blades and a reasonable cutter-life between sharpenings.

3. *Slot Milling with Slot Drill* also gives no difficulty in machining. Sample of slot-drilling on view. Time taken, 6 minutes. Size, 1¼ inches long, ½ inch wide by 1½ inches deep.

4. *Slot Milling with Slotting Cutter* is exactly the same as slab milling, and the same remarks hold good.

5. *Screw Milling* gives excellent results. The sample shown was cut with a multiple cutter 2⅜ inches diameter, running 60 revolutions per minute, work speed 1/6 R.P.M. Time taken, 6 minutes.

Drilling.—The question of drilling stainless steel has probably given rise to more discussion and more trouble than any other operation in the machine shops, most probably due to expecting the same results as obtained in mild steel. With the "F.G." in any condition up to and including 55/60 tons, no difficulty need arise if we reduce the speed by 10 to 20 per cent and the feed by at least 50 per cent compared with those used in drilling mild steel. For production work I would suggest a feed not exceeding 2¼ inches per minute, with speed of 400 revolutions for a ⅝-inch drill, whereas for mild steel the speed would be 420 to 450 revolutions and a feed of from 5 inches to 6 inches per minute.

There is just one more point I would like to touch on, and that is grinding and polishing. With "F.G." steel in the soft condition we have not got a satisfactory alundum or carborundum wheel yet. All grades we have tried leave a draggy surface, but when hardened up to the unmachinable condition the range of grades is considerable, but for a real good finish a soft grade is preferred. We use Aloxite Grit 30 Grade B.O., and (or) B.N., and for a mirror finish an X.F.I. Norton Wheel.

For polishing it will usually be found that No. 80 grade emery, mounted on the ordinary leather-faced buff, is a satisfactory medium for commencing the glazing operations. Should very bad surfaces be encountered it will probably be necessary to use a coarser grade, or regrind to a better surface. After using No. 80, and thoroughly bottoming all the irregularities, No. 120 grade emery may be used, mounted on a leather-faced buff in a similar fashion. After this stage some polishers prefer to follow with flour emery similarly mounted, whilst others go direct to the use of compos. These compos generally are stearine, carrying either emery alumina or chromic oxide, sometimes referred to as green compo.

All the tests except the planing have been carried out with tools made from ordinary "Speedicut" tool steel. This has been done to demonstrate that there is no real difficulty in machining stainless steel. Considerably better results can be got by using tools made from maximum "Speedicut," principally in the turning and drilling operations. The saving in time lost by grinding and re-setting of tools alone should more than repay the extra cost without taking into account increased production.—"Mechanical World," January 7, 1927.

THE BAULINO CONTINUOUS FUEL INDICATOR.

A practical difficulty in the care and running of multi-cylinder Diesel engines, both at sea and in land installations, is to ensure that each of the individual cylinders is carrying its fair and equal share of the load. The importance of this fact is well known to all those who have charge of large powered installations, and any simple and reliable method which will assist the engineer to ensure equality of supply should be welcome.

In modern engines of large power it is generally the practice to fit a separate fuel pump to supply the fuel into each cylinder, whether compressed air or airless injection is adopted; the only well-known exception is that which is called the "common or pressure rail" system. The quantity of fuel supplied by the separate pumps is generally controlled by the timing and period of lift of the pump suction valves, and adjustments are made on the tappets of these small valves, or on the mechanism which determines the point at which a "spill valve" opens; in either case no direct knowledge or measurement of the quantity of fuel actually injected is available or possible, except under research or test-bed conditions, and the driver is only able to gauge the effect of his adjustments by indirect means, such as indicator cards, test-cocks for observing the smokiness or otherwise of the exhaust, exhaust gas pyrometers, etc.

The adjustment of the fuel pump suction valve tappets is, shortly, a matter of trial and error, and can quite readily be made in the wrong direction, and may upset existing equality of fuel quantities injected, in the endeavor to equalize other conditions whose irregularity is not due to unequal fuel quantities at all, in spite of the symptoms observed. In trials of a 2000-horsepower, eight-cylinder submarine engine recently, which will be referred to later, in which the quantity of fuel to each cylinder was exactly known and controlled by the apparatus to be described, certain test cocks showed smoke, even when that cylinder was normally loaded.

In view of the development that is taking place in the use of large internal-combustion engines on land and afloat, there is thus an obvious need for some practical and simple source of information, in the light of which the engineer in charge can readily observe at any moment the amount of fuel being delivered to each or all cylinders and make adjustments accordingly and with confidence.

A solution to this problem has recently been brought out by Colonel Carlo Baulino, of the Italian Royal Naval Engineers, and it has been adopted by the Italian Navy for use on submarine engines. The apparatus contains no moving parts and depends on the application of known principles of hydro-dynamics, as will be explained later.

A small main tank A—Figure 1—of rectangular section is kept supplied with fuel from the ready-for-service tank, and the fuel is maintained at an approximately constant level, either by means of a float-controlled valve as shown, or in a later design by fitting a small air-cock in the closed top of the tank; this cock is normally kept closed, and the fuel level is kept sufficiently constant by the air entrapped above the level of the liquid, since that air compresses until its pressure balances the head of fuel from the service tank, in the engine-room.

From this common tank the fuel flows by difference of level into small subsidiary tanks C placed at the side of the common tank, passing through circular accurately made outlet orifices D. There are as many subsidiary small tanks as there are fuel pumps and cylinders; each pump draws its supply from a separate small tank C, and arrangements are made for any slight pump gland leakage, etc., to be piped back into its own tank C.

The orifices are all exactly equal in every respect, and the diameter is chosen having regard to the density and viscosity of the oil fuel and to the quantity that must pass per unit of time, so as to produce a fairly considerable difference of level between the common main tank and each of the subsidiary small tanks. Thus, if h is this difference of level, W is the area of the orifice, and Q the volume of liquid passing; then from the laws of hydro-dynamics we find that

$$Q = KW\sqrt{h}, \text{ or } h = \frac{Q^2}{(KW)^2}$$

where K is a constant depending upon the various magnitudes involved.

Since W is also a constant for any given size of orifice, we may write the equation as

$$h = C\,Q^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where C is a combined constant, and from this it is clear that the difference of level between the main and subsidiary tanks is proportional to the square of the volume of fuel delivered per unit of time; hence a very small change of the rate of flow will produce a large change of difference of level h.

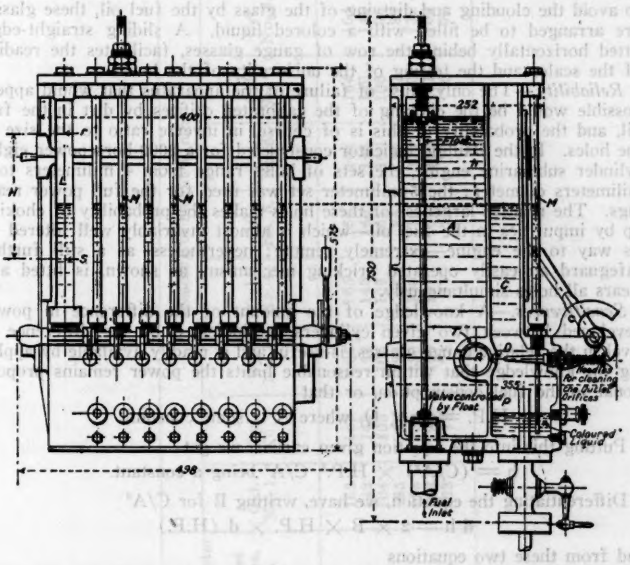


FIGURE 1.—ELEVATION AND SECTION OF BAULINO FUEL INDICATOR.

Floats, or other means of indicating the levels, are fitted both to the common tank and to each small tank, and to some scale the difference of level h between the common tank A and any one of the small tanks C shows the amount of fuel actually passing to the corresponding cylinder of the engine. Further, when the indicators on the small tanks all stand exactly at the

same height, the fuel supply to all cylinders must be exactly equal; if, however, one of these levels is above or below that of the others, the delivery of the corresponding fuel pump is less or greater respectively than that of the other pumps.

Now the difference of level h, besides depending upon the rate of flow Q and upon the diameter of the orifice, is proportional to the viscosity of the fuel, hence in the interest of a convenient and open scale for h it is desirable to be able to change the set of orifices, without undue trouble, to another set of larger or smaller size as the fuel is thicker or more fluid. This is arranged for by making several sets of orifices R in axial rows in a common cylindrical sleeve T; each orifice in any one row being exactly the same as any other in the same row, but the holes in each row being progressively of larger diameter; a partial rotation of the sleeve by the lever shown changes the whole set of orifices in use to a larger or smaller set, as desired, without stopping the engine, so as to obtain the most convenient scale for h to suit the grade of fuel being used.

The use of floats carrying pointers in each tank has, in practice, certain objections, such as bulkiness and guide friction; a set of gauge glasses is preferable, one for each cylinder M and one for the main tank L; in order to avoid the clouding and dirtying of the glass by the fuel oil, these glasses are arranged to be filled with a colored liquid. A sliding straight-edge, fitted horizontally behind the row of gauge glasses, facilitates the reading of the scales and the testing of the uniformity of the levels.

Reliability.—The only cause of failure of the apparatus that would appear possible would be the choking of the calibrated orifices by dirt in the fuel oil, and the probability of this is of course, in inverse ratio to the size of the holes. In the Baulino indicator constructed for a 2000-horsepower eight-cylinder submarine engine, the sets of holes range from 4 milimeters to 8 milimeters diameter; the 5-milimeter set was used for the full power readings. The relative largeness of these holes makes the probability of choking up by impurities in the fuel oil—which is almost invariably well filtered on its way to the engine—extremely remote; nevertheless, as a still further safeguard an easily operated pricking mechanism, as shown, is fitted and clears all holes simultaneously.

Sensitiveness.—A knowledge of the amount of the difference in power developed between two given cylinders, as indicated by the difference in level in the relative gauge glasses, is useful and is readily available by applying the knowledge that within reasonable limits the power remains proportional to the fuel consumption, or that

$$\text{H.P.} = A \times Q, \text{ where A is some constant.}$$

Putting this into the equation given earlier, we get

$$h = (C/A^2) \times \text{H.P.}^2, \; C/A^2 \text{ being a constant.}$$

Differentiating the equation, we have, writing B for C/A^2

$$d\,h = 2 \times B \times \text{H.P.} \times d\,(\text{H.P.})$$

and from these two equations

$$d\,(\text{H.P.}) = \frac{1}{2}\,\text{H. P.}\,\frac{dh}{h} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Taking as an example figures obtained on the submarine engine mentioned above and the tests of which will be referred to later, where H.P. = 250 per cylinder, h = 160 milimeters, and assuming a variation of 1 milimeter in the height of the level h, which is quite readable, we get from equation

Brake H.P.	Diameter of outlet orifice. mm	Reading of height h, mm.									Maximum variation.		
		Cyl. No. 1.	Cyl. No. 2.	Cyl. No. 3.	Cyl. No. 4.	Cyl. No. 5.	Cyl. No. 6.	Cyl. No. 7.	Cyl. No. 8.	Mean h, mm.	Of h in mm. dh.	Of power of one cylinder calculated by the formula $dP=\frac{1}{2}P\frac{dh}{h}$	Of power in one cylinder
												H.P.	Per cent.
2200	5	162	163	165	165	166	164	164	164	164	4	3·40	1·22
2005	5	142	142	144	143	145	144	146	146	144	4	3·50	1·40
1512	5	104	105	105	105	105	106	106	105	105·1	2	1·80	0·95
893	5	85	86	86	87	86	87	87	87	86·4	2	1·45	1·15
481	4	128	129	130	130	130	131	130	132	130	4	0·92	1·55
311	4	98	98	100	101	101	102	101	102	100·4	4	0·76	1·95
110	4	73	73	75	77	76	75	76	75	75	4	0·38	2·76

(2) that d (H.P.) = 0.8 H.P.; in other words, a difference of power between cylinders of under 1 horsepower can be easily detected in 250 horsepower by the indicator.

While the indicator is in process of manufacture, two marks are put behind the main tank gauge L at a vertical distance of H apart, corresponding to a known quantity of fuel, by accurate volumetric calibration; by using these marks the total consumption per hour of the engine can be readily taken at any time, by closing the fuel supply cock to the main tank and by taking with a stopwatch the time for the fuel level to fall between the calibrated marks; a correction being then applied for temperature or specific gravity of the fuel.

This simple operation, if performed as a matter of routine regularly at intervals throughout the twenty-four hours, is a very valuable check on the performance of the engine as a whole, either at sea or in a power station.

Practical Results.—Figure 2 shows the Baulino Continuous Fuel Indicator as made by the Fiat Company, of Turin, and applied by it to the 2000-horsepower submarine engines that it is building for the Royal Italian Navy. The bench test of the first of these engines was carried out for twelve days, running 300 hours continuously; during this time both the Baulino indicator and the usual calibrated test-shop fuel tanks were used simultaneously for purposes of measurement and comparison. The engine developed 2200 horsepower at 400 revolutions per minute for 76 hours, the normal full load of 2000 horsepower at 380 revolutions per minute for 212 hours, and for the remainder of the time a range of smaller powers at varying speeds. At the lowest power, 80 horsepower at the lowest speed 100 revolutions per minute, the regularity of the engine was remarkable, and this result was facilitated by the adjustment of the fuel pump tappets in such a way as to maintain the whole of the eight levels in the gauge glasses at a constant and uniform height, thus ensuring that each and every cylinder was receiving the same quantity of fuel and was equally loaded.

No obstruction of the calibrated orifices was experienced throughout the long trial, and from the appended table of some of the observations made, it will be noted that the maximum deviation in power among the eight cylinders, calculated by the formula (2), was 3.4 horsepower at the highest power of 2200 horsepower, or on an average power of 275 horsepower per cylinder this amounts to 1.22 per cent.

Measurements of the total fuel consumption of the engine at this load taken by the indicator or by the shop tanks were 194 and 195 grammes per horsepower hour, respectively, and *pro rata* at the other powers.

The indicator is installed close to the fuel pumps or on the control platform, and the information it gives is thus continuously under the eye of the engineer, and has been found to be unaffected by the movement of the ship in a rough sea.—"The Engineer," February 11, 1927.

ANCHOR BOLTS FOR REFRACTORIES.

The use of oil as a fuel in naval vessels about 1910 introduced new problems in furnace design, one of the most difficult ones being furnace refractory linings and insulation. The higher temperatures obtained in the oil fired boilers required the development of better refractory brick. This part of the problem became more difficult because of the fact that very large changes in furnace temperatures had to be provided for and also because of the possibility of maintaining high furnace temperatures for prolonged periods by the use of oil.

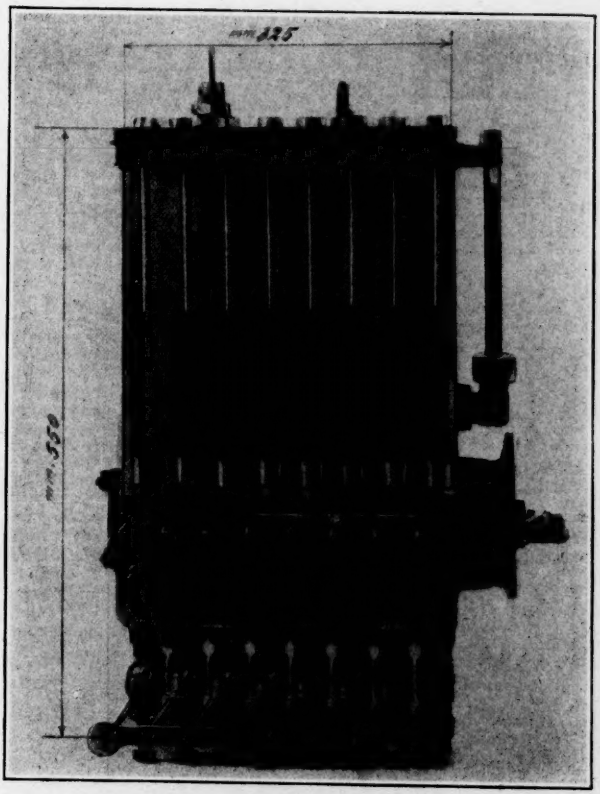


FIGURE 2.—FRONT VIEW OF INDICATOR.

Anchoring of the refractory brick also became a more important and difficult problem in oil fired furnaces not only because of the higher temperatures, but also because of the large areas of furnace linings which were subjected to "panting," which in the early days of oil burning was not at all an uncommon condition at high rates of combustion.

The first refractory brick used was a special brick about 9 by 9 by 2. Each brick was held in place by one anchor bolt as shown in Figure 1. The head of the bolt fitted into a formed recess on the fire side of the brick. After building the walls, these recesses or holes were filled up with a plastic cement or fire clay to protect the head of the bolt from direct contact with the furnace temperature. It was, of course, very difficult, if not impossible,

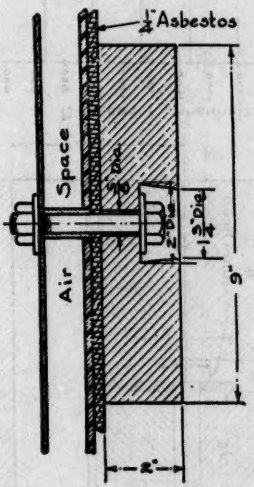


FIGURE 1.

to plug up these holes with a suitable material that would not shrink away from the brick or break up and fall out due to the changes in temperature. As a result the heads of the bolts were continually burning off, and after any sort of steaming it was usually necessary to do considerable rebricking, or at least to replace many of the anchor bolts, large supplies of which were always carried on board.

It was not long, of course, until both the form of brick and the method of anchoring were changed. The next brick used was of a commercial size—that is 9 by 4½ by 2½—and was anchored as shown in Figure 2.

With the brick anchored as shown there was still considerable difficulty experienced with the anchor bolts and an attempt was made to cool the anchor bolts by beveling the corners of the brick below the anchor bolts, as shown in Figure 2. At the back of this space formed by the beveled edges of two adjacent bricks a small hole was drilled through the casing so

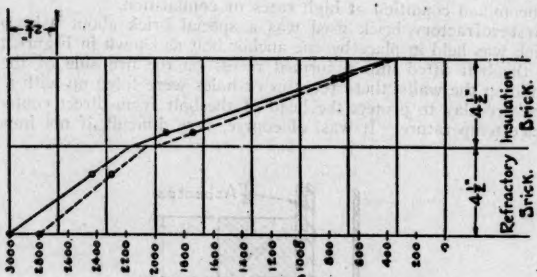


FIGURE 3.

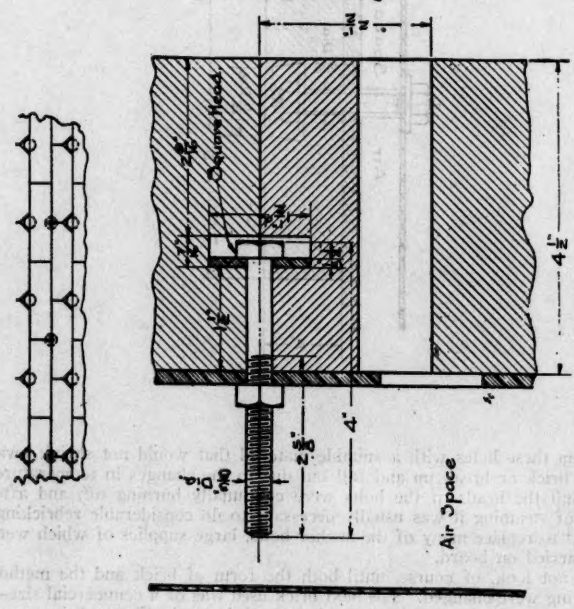


FIGURE 2.

that air from the fireroom could enter the furnace below the bolt and thus cool it. It is needless to say that this method was soon abandoned. Various other attempts were made to air cool the anchor bolts by hollow shanks, radiating fins, etc., but without success.

In order to reduce weights to a minimum, particularly on the lighter, higher speed ships, both efficiency and comfort were sacrificed and it was customary to insulate oil furnaces by one course of refractory brick 4½ inches thick, back of which and inside the furnace casing was usually placed ¼ inch or ½ inch of asbestos mill board. With practically no insulation back of the brick the temperature gradient through the brick was of course quite steep, and the anchor bolt heads being protected by two and one-half or three inches of the refractory were not exposed to the extremely high temperatures of the furnace. It was therefore not very difficult to find a material that would produce fairly satisfactory results. At least an anchor bolt material could be found that would have the same life as the bricks.

Inasmuch as strength of the material was not a requirement (the strength of the refractory being the limiting factor in this connection) it was necessary only to find a material that would not oxidize or become too brittle at the temperature to which it must be subjected. Many materials were tested and taking into consideration costs, ease of manufacture, and performance, anchor bolts of monel metal either cast or rolled were found to be the most satisfactory.

Plugged holes similar to those used in the earlier brick and described above were of course abandoned and every effort was made to keep the bolt head as far away from the furnace face as possible and to protect it as much as possible by the refractory. The type of brick which was used quite successfully and adopted as a standard was similar to that shown in Figure 2 except that the corners were not beveled.

In naval vessels, oil is burned by what is known as the mechanical atomization method and the closed fire room system of forced draft is used. When a fire room is being operated at fairly high rates, a considerable amount of air is forced through the fire room and into the furnaces. As a result, the temperature of the fire room is quite reasonable. At low powers and while at anchor, however, such is not the case and unless the forced draft blowers are operated at capacities higher than actually necessary to furnish the air required for combustion, temperatures in the fire room where the furnaces are poorly insulated become excessive. The result is that the fire room personnel operate the blowers at higher capacities than necessary to keep the temperatures down. This of course is a very uneconomical practice.

In order to provide more comfortable conditions for operating personnel and to increase fire room efficiency by decreasing furnace radiation and to reduce somewhat the tendency of personnel to operate the large forced draft blowers at excessive speeds, more recent practice, in practically all types of ships, has been to provide considerably more insulation. The present specifications now require a 4½-inch refractory wall back of which is a 4½-inch insulating wall of calcined diatomaceous earth brick.

This increased insulation makes the anchor bolt problem quite a different one. The temperature gradient through the refractory brick is no longer so steep and the head of the anchor bolt which is embedded in the refractory is now subjected to very high temperatures.

When destroyer furnaces were first insulated in the above manner it was realized that the anchor bolts would be subjected to much more severe temperatures and that the monel bolts, then used as standard, were not entirely satisfactory. However, at that time a suitable material had not been found and the first few installations were made with monel bolts.

Results were even worse than had been expected. After only short periods of high power steaming the furnace walls began falling down, due to failure of the anchor bolts. In the meantime, tests of many different kinds of materials were being made at the Fuel Oil Testing Plant at the Philadelphia Navy Yard, in order to obtain an anchor bolt which would withstand the necessary high temperatures and which could be manufactured at a reasonable price. Bolts made from these materials were actually installed in a furnace and subjected to furnace temperatures ranging from about 2800 degrees to 3000 degrees Fahrenheit for periods of 24 to 30 hours. The actual temperatures at the bolt heads ranged from 2200 degrees to 2550 degrees Fahrenheit. Figure 3 represents the approximate temperature gradient through refractory and insulating walls with the above furnace temperatures.

Some of the materials tested and the results of the tests are as follows:

A. Class "B" steel. This material fused at low temperatures. The fused metal cut deeply into the brick and actually carried down the furnace brick work in places.

B. Class "B" steel specially treated to prevent oxidization at high temperature. This material was not appreciably better than the untreated material.

C. Rolled Monel. This did not fuse but oxidized excessively. The bolt "puffed up" and became very brittle.

D. Cast Monel. In some cases this appeared slightly better than the rolled monel but in other cases not so good. This difference in performance was undoubtedly due to the differences in foundry practice in casting the metal.

E. Special alloy of following approximate chemical composition:

Nickel, per cent	35 to 36
Chromium, per cent	3 to 3.5
Manganese, per cent	2
Copper	remainder

The performance of this material was very similar to that of the rolled monel.

F. Special alloy of the following approximate chemical composition:

Nickel, per cent	72.8
Chromium, per cent	19.8
Iron, per cent	3.6
Manganese, per cent	2.2
Copper, per cent	1.6

This material performed very satisfactorily. It oxidized only slightly and retained its strength.

*G. Manganese-nickel, approximately 96 per cent nickel and 4 per cent manganese. This material was very satisfactory. In test it became coated with a very thin film of manganese oxide and was very tough after test.

* Class "G" alloy covered by U. S. N. Specif. 46-N-3, dated February 1, 1927.

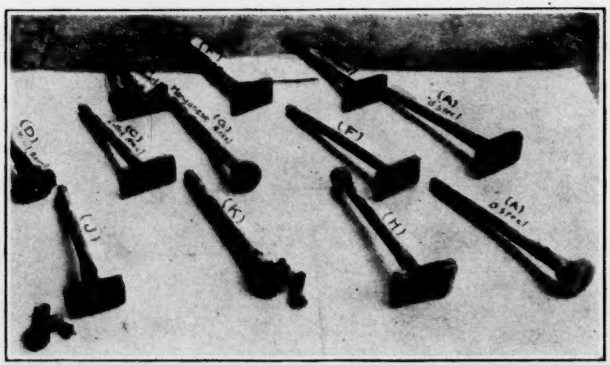


FIGURE 4.

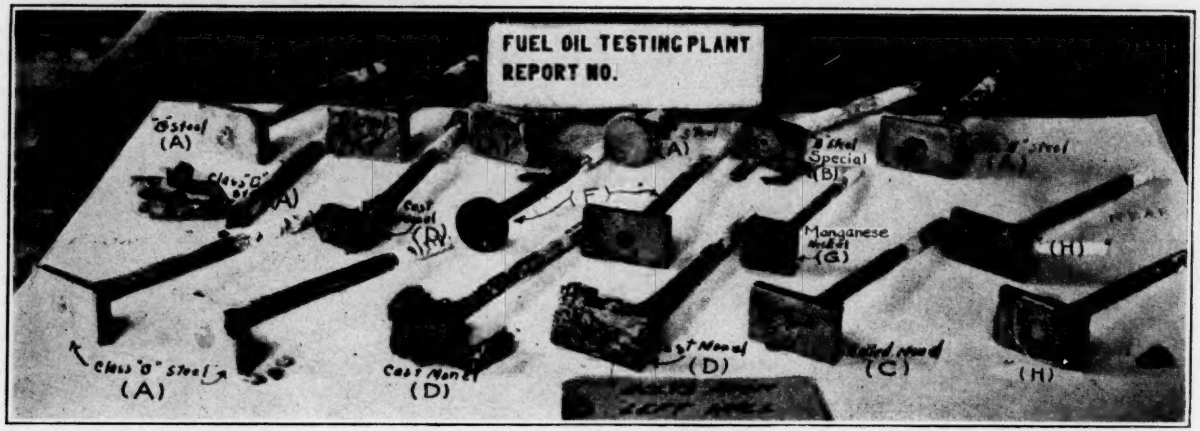


FIGURE 5.

H. Special alloy of following approximate chemical composition:

Chromium, per cent	16.50 to 18.50
Iron, per cent	81.60 to 82.50
Manganese, per cent	.30 to .50
Silicon, per cent	.50 to 1.25
Carbon, per cent, under	.10
Sulphur, per cent	.04
Phosphorus, per cent	.04

Performance very similar to that of Class "B" steel.

I. Special alloy containing 27 to 30 per cent chromium, the balance iron with small amounts of sulphur, silicon, manganese, and phosphorus.

This material oxidized only slightly but became very brittle.

J. Special alloy of following chemical composition:

Chromium, per cent	12 to 16.
Iron, approximately, per cent	85.
Manganese, per cent	.50
Silicon, per cent	.50
Carbon, per cent, under	.12
Sulphur, per cent	.025
Phosphorus, per cent	.025

Performance very similar to "I." Bolts of this material became very brittle under the head.

K. Special alloy. Chemical composition:

Chromium, per cent	20.
Iron, per cent	78 to 79.
Copper, per cent	1.
Carbon, approximately, per cent	.30

Oxidized heavily, and performed in general very similar to Class "B" steel.

Of the above materials only "F" and "G" were considered as satisfactory for use in naval furnaces. The special alloy "F" due to its excessive cost was discarded in favor of "G," the manganese-nickel (4 per cent manganese). Subsequent tests were made of a manganese-nickel bolt containing about 5 per cent manganese. This material proved to be equally satisfactory as the 4 per cent manganese. The accompanying photographs (Figures 4 and 5) show the condition of bolts of some of the above materials upon completion of the tests.

As a result of the above tests a standard manganese-nickel (about 4 per cent manganese) anchor bolt, as shown in Figure 6, has been adopted for use in the furnaces of Naval Boilers. This sketch also shows the standard fire brick and the method of anchoring. Both the bolt and the method of anchoring have proved to be quite satisfactory in actual service in naval vessels.

In connection with the manufacture of manganese-nickel anchor bolts, the following comments on both the cast and forged bolt, may be of interest:

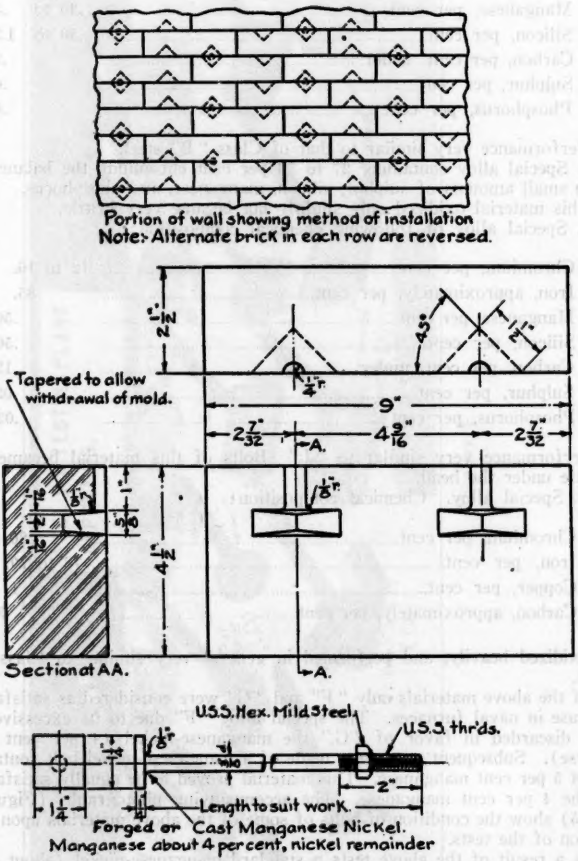

Portion of wall showing method of installation
Note: Alternate brick in each row are reversed.
Tapered to allow withdrawal of mold.
9"
A.
Section at AA.
U.S.S. Hex. Mild Steel
U.S.S. thr'ds.
2"
Length to suit work.
Forged or Cast Manganese Nickel.
Manganese about 4 per cent, nickel remainder

FIGURE 6.

Cast bolts, that is the finished bolts, should conform to the following chemical analysis:

Nickel, minimum, per cent		93.0
Manganese, per cent	3.50 to	5.
Iron, maximum, per cent		2.0
Silicon, maximum, per cent		.50
Carbon, per cent	.05 to	.30
Sulphur, maximum, per cent		.035

Bolts having the minimum impurities will bend through 180 degrees without fracture, while those having the maximum impurities permitted will bend through 90 degrees, which is considered satisfactory.

In order to produce uniformly successful results, "reverts" should be kept low, not more than 25 per cent.

Various methods of foundry practices were used when the manganese-nickel bolts were first considered. It soon developed however that better bolts could be manufactured at less cost by forging. Most navy yards forge the bolts from ⅜-inch stock, upsetting the heads in three steps. One yard is at present experimenting with ⅝-inch stock, forging the shank to ⅜ inch and upsetting the head in two steps. A sufficient number of bolts have not been manufactured in this manner to definitely show any advantage from the use of ⅝-inch stock.—"Pacific Marine Review," March, 1927.

NIGHT FLYING IN BAD WEATHER.

By Edmund T. Allen.

Across the 500-mile course from the plains of Nebraska to the Utah Salt Lake desert, air mail pilots have to encounter practically all conditions of flying. From a minimum of 4200 feet above sea level, where the worst flying conditions consist of a fog and smoke-covered terminal city valley, they fly to the higher ranges of the Continental Divide where snow storms and fog reach to heights greater than the ceiling of the mail plane. In daylight these conditions are often well-nigh heart-breaking to the pilot; at night they are many-fold worse. It is a striking fact that each offers a brand new problem, one which must be solved on its own merits without reference—except for the accumulation of general experience—to those of the day before.

Flying at night back and forth across the chief ranges of the Rocky Mountains regularly for some years has precipitated certain outstanding problems in the minds of the pilots of the Trans-Continental Air Mail Service. Chief among these are, first, the great difference between day and night flying; second, between flying over a familiar and an unfamiliar course; and, third, and by all odds the greatest distinction to be made, that between good weather night flying and bad weather night flying.

It is only the last of these which will be discussed here, and we shall try to outline, in general, the problems involved and some of the ways in which they may be expected to be solved in the near future.

The lighted airway, of which we are all so proud, is a magnificent gesture in the direction of providing an adequate aid to the pilots flying at night between New York City and Salt Lake City. It consists of revolving lighthouse beacons placed every twenty-five miles (in the East considerably

closer) in practically a straight line across the continent. It is the intention of the Department of Commerce to continue this string of lighthouses during the Summer of 1927 all the way to San Francisco. One of the pilots of the Trans-Continental route, who, the records show, has flown more miles at night than any other man in the world, is given credit for the remark, now proverbial, that the present lighted airway is not needed in good weather and is no good in bad weather because one cannot see the lights through fog and snow. This is, of course, only partially true, for there are many half-way-between nights when the absence of the beacons would surely delay the mail.

Nevertheless, the whole question of the lighted airway needs to be reconsidered from this point of view. On one clear night the writer has counted eighteen of these twenty-five-mile beacons at one time when flying at 16,000 feet over the Rockies! Their absence on such a night would make no difference whatever to the ease of flying mail or passengers across the continent. On many another night when fog and snow are thick, one may find the only possible course lies at fifty feet above the Union Pacific tracks or the Lincoln Highway, or beside a barb-wire fence which one knows by heart. Under such circumstances we may pass within a mile of one of these powerful beacons without ever seeing it.

The problems which the clear night offers are outside the scope of this paper. Those of the bad weather may be considered under five headings: first, navigation with reference to three perpendicular axes, *i.e.*, keeping the airplane flying level and straight; second, navigation with reference to a course between destinations; third, landing where the field is obscured; fourth, engine operation; and fifth, the collection of ice on wings and wires. The most pressing of these are the first two, namely, blind flying, and the preparation of a lighted airway which can be used in fog and snow. To these two we shall limit ourselves in the present discussion.

There is a good deal being written about the navigation of airplanes under conditions of "zero visibility," when the ground, horizon, sun, moon or stars are not visible. In such weather, flying must be done either by instruments or by so-called "feel." While it is quite certain that such flying can be accomplished for short periods in relatively smooth air, it has never been positively demonstrated as possible for an indefinite time. The point is that very often reference points, such as glimpses of the ground or a star in the mist, or the aura of the moon through clouds, or a dim dark line of horizon in a gray sphere encompassing the machine, become visible momentarily or constantly without the consciousness of the pilot that such is the case.

Many aeronautical people wonder why complete blind-flying is so difficult. The explanation is that the acceleration of a turn, the acceleration imparted to the airplane by gusts, and the acceleration of gravity are mechanically indistinguishable from one another and cannot be separated by an instrument or by "feel." The pilot knows what the resultant of these accelerations is but he is at a loss to find the direction of the gravity component. In an airplane especially fitted up to simulate blind-flying by completely enveloping the pilot in a covered cockpit, one air organization endeavored to develop blind-fliers with all the instruments at the technician's disposal. The result was comical—only escaping the tragic by the provision of a reserve pilot in the rear cockpit of the airplane who always recovered the machine from the spin, slip, or spiral dive into which it seemed inevitably to go at the hands of the "blind" flier. It is impossible to say whether or not an indefinite amount of practice would lead to a great improvement in skill. One of the pilots who had had most practice in the blind-flying cockpit, when

caught in heavy cloud at night soon after his practice, promptly went into a spin and recovered only after the plane had emerged below the storm, 400 feet from the ground!

How shall we solve this problem? Some research carried out by one of the larger instrument makers resulted in the conclusion that an absolute horizon-indicator was an impossible thing to build even though twelve gyroscopes were used, four for each of the three axes perpendicular to each other. Is it, then, impossible to "fly fog?" We think it is not impossible; but the solution of the problem will lie in a combination of our present turn indicator and a more stable airplane which will fly itself, laterally and longitudinally. It may be necessary to have two turn indicators, one with a coarse adjustment for very bumpy air and for getting out of difficult positions, and one with a very fine sensitivity to be used in flying a given course in connection with the compass. These, together with a stable airplane, would seem to offer the most hopeful possibilities for ultimately successful blind-flying.

In the meantime pilots who are flying at night in had weather keep within sight of the ground even though it means maneuvering down canyons or following broken cliffs or irregular divides within a few feet of the ground. It is their only means of safety when they know their course intimately.

The second difficulty in bad weather night flying is in keeping on a course or in keeping from getting lost when one tries to take alternative courses. As was said before, powerful revolving beacons on the lighthouses at 25-mile intervals are of relatively little use when the air is thick with snow, rain or fog, or a combination of them. In such conditions, if one flies at all, it must be over terrain with which one is very familiar. Flying through mountain defiles, when the ceiling and visibility are extremely poor, is little short of suicide unless the pilot knows the terrain almost like the blind man knows his familiar haunts. Even though one knows the mountains from 5000 feet above "like a book," they become a foreign land when one is down between their cliffs.

There are three passes over the fog-covered Wasatch Range into the Salt Lake Valley. When the usual two are impossible to get through, the pilots of the mountain division go forty miles north of the course to Devil's Slide and Weber Canyon whose precipitous sides are barely wide enough to clear the wing tips. Very often the fog is half way down in Weber Canyon and if the mail is to get through, the airplane must travel in the lower third of this defile. The pilot who is not familiar with every promontory here does well to avoid entering Weber Canyon, for turning around, once one has entered, is out of the question.

FOLLOWING THE RAILROAD.

As a rule, the flying on this division as well as on the entire route is over relatively open but heavily broken country where landing outside of prepared fields spells disaster. Here the railroad is the alternative course which it is possible to follow when the beacons cannot penetrate the fog. Under these conditions the wing-tip lights on the airplane are of little use. In snow or fog-filled air, the diffusion of the beam reflects back into the pilot's eyes and blinds worse than it illumines the way. The stronger the beam, the more it seems to blind one, and the usual procedure is to turn off all lights and get one's eyes adapted to darkness so that one can see even slight irregularities in a dark snow-covered landscape. Very often the roadbed of the railway is the only discriminable object in the absolute gray of land and sky. Then, when flying at 120 miles per hour with one wing fifty feet above

the tracks, often a turn of the rails will put them out of sight under the fuselage. If one cannot quickly enough skid aside to reveal them again, one has the unwelcome opportunity of trying one's blind-flying ability. In one such case when the railroad unexpectedly entered a tunnel, the pilot crashed against the mountain-side before he could pick up his landmark on the other side.

It is not to be assumed that this sort of flying is the regular course of events or that the pilot who emerges from such a flight does so with any other feeling than wondering a little that he is still alive. Otherwise the air mail casualty list would be much larger than it is. There are a considerable number of cases on record of pilots touching the ground in fog without knowing they were so near to it. Such incidents practically always result in complete crashes. Usually, however, the pilot returns to the nearest field to await an improvement in visibility, or he is warned by signals of bad weather ahead and, cautiously, does not enter it. With the added responsibility of passenger carrying soon to be entered upon, this caution must be increased many-fold.

There are very often, however, flights in which the pilot will be caught in the bad weather and find it impossible to turn around without danger of losing sight of the ground or of his landmark, or he may feel that he would rather go on than go back through what he has just flown, on the assumption that what is ahead cannot possibly be as bad as what is behind. It is only in such cases of being caught in the bad weather and finding it impossible to land or get back to the last emergency field, or in cases when all the emergency fields are snowbound and unusable, or where one believes that the very bad visibility does not extend for many miles along the course, that such flying is done at present. It is, however, surprising to find the large amount of this sort of flying that is going on, on the Trans-Continental route. If it were not done at all, the 94 per cent record of the Government air mail would be very considerably lower than it is.

How, then, shall we meet the problems of "zero visibility" in the future? Some talk of the radio beacon with its clear indication to the pilot when he is on or off his course. But the radio beacon almost of necessity implies blind-flying facilities, and here we meet with our first problem which must be solved before we can attempt blind-navigation. There are a few cases, such as flying *over* fog, clouds, or storms, in which the radio beacon would assist without the necessity of blind-flying, but these cases often involve the third problem of landing in fog-bound terminals, a problem somewhat farther from solution than either of the others. In case of radio communication with the pilot he can be informed of the state of the terminal field and of the weather on each side of the terminal. If a clear field is found, one can perhaps fly to it, providing it remains clear until it is reached, and providing engine trouble does not develop, in which cases the blind-landing is again to be faced. There is no doubt, however, that a combination of radio beacon, radio communication, and excellent weather information will enable an improvement to be made in the number of completed bad weather trips.

We have, however, still not solved the problem. There would be many nights when flying would be possible near the ground *if the ground were illuminated,* when it would not be possible even though radio beacons work perfectly, until blind-landing has reached a state of perfection hardly imaginable today. This being the case, we must look toward a replacement of our present lighted airway with a system of lights that will be serviceable in bad weather. I would here propose the following system. It may be that it will involve a greater expenditure than the present string of lighthouses, but some day, when the amount of air travel warrants the change or the

public demands it for the increased certainty of air mail or passenger service, something of the kind will enable pilots to achieve a 100 per cent efficiency in night air travel.

I would propose the substitution of a series of closely set stationary arc lights for our present revolving beacons. Every pilot knows how relatively easy it is to fly over a lighted town or a long street with arc lights a block apart even when fog makes flying over the unlighted country or mountain sides impossible. If such a series of lights, set about 300 feet apart in a line preferably along a straight road but, alternatively, straight across the great open spaces, were arranged so that the lights could be operated and controlled from central high-voltage power stations located every twenty-five miles in the same places as the present lighthouse beacons, then the present caretakers, with somewhat larger power units, could manage them at a very small expense, especially since they would need to be turned on only while the airplane was passing over that particular sector. On clear nights they could be left off altogether. The lights themselves should have wide reflectors directed upward, and should be set on short poles just high enough to enable an airplane clearing them to clear also any obstruction between them.

Such a system of lights would enable flying to be continued where now it must be defaulted. A light every 300 feet (when the people can afford it they could be put even closer together and might serve to light a Trans-Continental highway) means to the pilot flying at 120 miles per hour that he will cross one every 1¾ seconds, a frequency that will enable him to check constantly his horizon and altitude.

The value of the enormous beacons of one billion candlepower or more such as are being installed on some air routes for lighting long stretches of the course, is distinctly questioned by those who have to do the flying over them. When they are most needed in heavy mist and fog they penetrate only a few miles; while, if the expense and power they represent were distributed over the course in a thousand small lights set sufficiently close together, flying could be continued many miles beyond the point to which even the most powerful single beacons penetrate.

It might be a long step in advance if the Department of Commerce which is now taking over the lighted airway, were to try out such a system of lights on a small sector in one of our worst fog belts. It is only by such means that we can hope for real improvement in night flying efficiency. If the recently announced nightly passenger service to be inaugurated from Chicago to San Francisco is to attain continually increasing efficiency and safety, it must look toward the solution of these two most pressing problems of bad weather night flying, namely, blind-flying and a satisfactory lighted airway.—"Aviation," March 7, 1927.

NOTE: The present acetylene blinker lights set at three-mile intervals serve admirably the purpose of marking intermediate fields, but as routing beacons they are of little value. When the large revolving beacons cannot be seen from station to station, the blinker lights are not visible much more than one hundred yards. They do not give a clear indication either of one's height above them or of a horizon line, and it is very easy to get lost between them.

ASSOCIATION NOTES.

CHANGE IN COUNCIL.

Captain S. M. Robinson, U. S. N., a member of the Council of the Society, has been detached from duty in Washington to assume duty at the Navy Yard, Puget Sound, Wash. Commander O. L. Cox, U. S. N., has been appointed to fill the unexpired term of Captain Robinson.

MEMBERSHIP

The following members have joined the Society since the publication of the last JOURNAL:

NAVAL.

Cotton, C. F., Lieutenant (CC), U. S. N.

Dennison, R. L., Lieutenant, U. S. N.

Keenan, Walter, Lieutenant, U. S. N. R., 210 W. 112th Street, New York, N. Y.

Okie, J. B., 71 Beekman Street, New York, N. Y.

CIVIL

Bliss, Edwin C., Central Manati, Oriente, Cuba.

Perrott, William, 961 Palisades Avenue, Woodcliffe, N. J.

Simpson, William K., 61 Fiske Street, Waterbury, Conn.

ASSOCIATE.

Brennan, Richard C., 1519 Railroad Avenue, South, Seattle, Wash.

Callis, William H., 141 W. 109th Street, New York, N. Y.

Israel, Philip N., Worthington Pump & Machinery Corporation, 426 Homer Building, Washington, D. C.

PRIZE ARTICLE ON ELECTRIC WELDING.

The Lincoln Electric Company of Cleveland, Ohio, in a sincere desire to improve the art of electric welding and to indicate the advantages and economies to be gained by its use, has made available the sum of $17,500 to be given as prizes for the best papers submitted on this subject.

The American Society of Mechanical Engineers has accepted custody of the fund and will conduct the competition. A committee of judges appointed by the Society will pass upon the relative merits of the three best papers presented. A prize of $10,000 is offered for the best paper submitted; $5,000 for the second best, and $2,500 for the third best.

The widest competition possible is desired. The rules of the competition and any information desired by those interested may be obtained by writing the American Society of Mechanical Engineers at 29 West 39th Street, New York, N. Y.